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AIRCRAFT ENGINE SUMP FIRE
MITIGATION, PHASE II

BY

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FOREWORD

The research described herein, conducted by the Technology Services Division of SKF Industries, Inc. was performed under NASA Contract NAS3-19436. The work was completed under the management of NASA Project Manager, Mr. William R. Loomis, Fluid System Components Division, NASA Lewis Research Center. Dr. G. Domoto of Columbia University consulted in the formulation and creation of the software in the analytical portion of this program.

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SUMMARY

The research performed on this program was conducted to investigate turbine engine bearing sump configurations; the objective being to obtain information which would suggest limits on the variables of oil flow rate, air flow rate, and input temperatures as well as geometrical variations which would reduce the probability of sump fires. The work performed consisted of both an experimental and analytical investigation.

The experimental work was performed on an existing NASA owned high speed bearing test rig designed and built by SKF Industries to simulate an aircraft engine mainshaft design and modified to accommodate the requirements of the fire study. The rig included seven thermocouples within the bearing sump to sense the oil-air mixture temperature and determine the presence of fires. Electrical spark ignitors were used as an easily applied, reproducible method of igniting fires when susceptible conditions existed. A baffle plate was located between the bearing and the simulated hot air leakage port to minimize the uncontrolled mixing of the oil and air, and to reduce the chances of excessive hot air producing a bearing failure.

A total of four test series was performed with each series evaluating the possibility of igniting fires within a given range of input variables. Test Series 1 was performed with the initial baffle plate design, and consisted of five runs incorporating the full range of selected input variable values. These were:

Oil inlet temperature -	353-467°K (175-380°F)
Oil flow rate -	0.23-0.46m ³ /hr. (1-2 gpm)
Hot air temperature entering hot air chamber -	622-833°K (660-1040°F)
Hot air flow rate through bearing sump -	7-48stdm ³ /hr. (4-28 scfm)

Test Series 2 was performed with a modified baffle designed to further minimize the oil and air mixing, and included approximately the same variable values used in Series 1. Test Series 3 and 4 were both performed with the modified baffle configuration, and all runs initiated with relatively low mixture temperatures in the sump, which were then increased during the test by increasing the inlet oil and/or air temperature.

In the analytical portion of the project, two basic mathematical techniques were successfully formulated and computerized to aid in the study of the flammability condition in the bearing sump. The first analysis program traces two phase oil flow (liquid and vapor) in an air stream passing through a cylindrical geometry. This was utilized to perform a parametric study to determine the influence of selected variable changes on the generation rate of oil vapor with respect to distance of travel in the tube. It was further utilized to perform a comparative evaluation with selected test data. The second analysis considers the ignition of the vapor-air mixture by an ignition source and allows the determination of combustion or its absence. This program was exercised with a two phase mixture of decane to demonstrate proper operation.

The experimental program demonstrated the following results:

1. Fires could be ignited over the full range of air and oil flow rates and air temperatures evaluated: Air flow, 6.8-49.3 std m³ (4-29 scfm), Oil flow, 0.23-0.57 m³/hr (1-2.5 gpm), Hot air temp. 622-868°K (660-1050°F).
2. No fires could be ignited when the oil inlet temperature was maintained below 417 K° (290°F).
3. The severity of the fires ignited were found to be directly proportional to the hot air flow rate.
4. Fires were readily ignited in many cases with high oil flow rates, but not with low oil flow rates even though the air flow rate and the air and oil temperatures were maintained at a constant value.
5. A reasonably good correlation (ability to ignite or not ignite fires) was found to exist between the mixture temperature at the ignitor and the calculated flammability limits as defined by flammability theory. This was especially true when the oil inlet temperature was above 432°K (320°F). This approach for determining flammability conditions within the sump is considered to be reasonably reliable especially if the residence time of the oil droplets in the hot air is considered to be long.

The analytical program produced the following results:

1. The parametric study performed with the computerized mathematical model showed that oil droplet size and air temperature had the greatest influence on the generation rate of oil vapor.

2. The correlation between the test data and the analytical data was shown to be good considering the assumptions that had to be made to compensate for the major differences existing in the rig geometry and the simplified geometry used to establish the mathematical model. Thus the analytical approach used is considered to be a viable method of determining flammability conditions within a bearing sump when values of the variables can be established.

I.O. INTRODUCTION

A recognized need exists in the aircraft gas turbine industry for establishing causes and conditions under which fires are ignited in the mainshaft bearing chambers and how these fires can be prevented or quickly extinguished.

Lubricant sump fires have been encountered in high temperature operation of aircraft engines during flight, in engine ground studies, and in advanced laboratory studies of lubrication systems (1)* and mainshaft seals (2). There is evidence that at least 31 incidents of sump fires or excessive heat in a bearing sump have occurred over a recent 5-year period in one widely used aircraft engine. Despite the reality of fires and near fires in operational aircraft engines, the high oil-recirculation rates used in engine sumps leads to the general contention that these areas are normally too oil rich for the initiation of fires. This contention is based on a false premise that the quantity of liquid oil is a primary variable in flammability. This is not the case. The concentration of oil vapor and the temperature determine flammability limits. However, the trend toward developing engines with higher speeds, higher pressure ratios, and resulting higher energy levels suggests an impending increase in the frequency of sump fires.

Past sump fires have resulted from a number of different causes under different sets of conditions. Due to the complexity of the problem, the initial task of the program was to investigate experimentally the possibility of establishing limits on the controllable input variables (oil and air temperatures and flow rates) to the sump over a range of values present in engines, which would produce flammability conditions within the sump. The second task was to extend and obtain a better understanding of the information obtained in Task 1 by performing analytical studies to show how the input variables react with each other to produce conditions that would be susceptible to fire ignition. The goal of the program was thus to obtain knowledge which would suggest limits on the input variables and modifications to the internal geometry of the sump which would minimize or eliminate sump fires.

1.1 Preliminary Experimental Study

Results from a preliminary experimental study performed several years ago by SKF Industries on NASA Contract NAS-3-14310 and reported in CR-121158 (3) had indicated that spontaneous combustion could not be obtained over the range of variables studied. However, simulated engine fires could readily occur and, in many cases, be self-sustaining over a wide range of

*Numbers in parentheses refer to List of References at the end of the report.

parameters when an ignition source, in this case an electric spark ignitor, was present. Other significant results were as follows:

Ignition from rubs by labyrinth seals and other component materials were shown to cause sump fires. Bearing skidding and excessive seal interferences are potential fire ignition sources and suggest that accidental fires in engine sumps may well arise from these causes.

Fire ignition is sensitive to location of ignition source and/or variations of mixture conditions within the sump area. It is likely that significant real differences in air-oil ratios exist in the various parts of the sump, making it difficult to achieve significant data on air-oil ratios. Oil degradation products indicated that sump fires begin in localized small regions of the sump and are thus influenced and controlled by baffles. Combustible volume grows slowly with the duration of the fire in response to local gas and oil mass flow conditions.

Nitrogen blanketing was effective in the immediate extinguishing of every test run fire once the fire had been detected.

A fire-baffle (Monel sheet) mitigation device on the hot side of the bearing not only prevented fire propagation, but also prevented bearing thermal seizure due to hot (922°K; 1200°F) gas flow directly into the bearing. Such baffles have practical significance.

Freon-113 flame snuffer injected into the lubricant flow was only marginally effective in controlling fires.

1.2 Flammability Background

Background subjects considered important in the study of sump fires include: 1) basic conditions necessary for fires to start, 2) the flammability limits for hydrocarbon lubricant vapors, 3) the importance of engine sump sealing systems, 4) and engine operating parameters that affect fires. A brief discussion of these subjects are presented here to provide a background and better understanding of the problem for the reader. Extensive information on the subjects of burning of hydrocarbons and their flammability limits are presented in references 4 and 5 which in turn list many additional references.

Three basic conditions are considered to be necessary in an aircraft engine oil sump for fires to occur. First of all, there must be adequate air and oil so a combustible mixture of oil vapor and air, or of fine oil mist and air can be formed. If there is insufficient oil vapor in the mixture (too lean) or excessive oil vapor in the mixture (too rich), a fire cannot start. Data from reference 6 show that for a MIL-L-7808 (type I ester) lubricant, fires cannot be ignited in static conditions if air-oil weight ratios are above 29 to 1 or below 5.5 to 1.

Secondly, the oil temperature must be above a critical value so sufficient oil vapors are formed to produce a combustible mixture or a self-sustaining flame (one which continues to burn in the absence of an ignition source). Under static conditions, these are referred to as the flash point and fire point. It is also possible that very fine mist of droplet sizes less than 10 microns will react as though it were an oil vapor with respect to fire ignition (6). At temperatures above the autoignition temperature (AIT), no external ignition source is required to start a fire. For the type II ester oil used in this program the flash point, fire point, and auto-ignition temperatures are 525°K (485°F), 558°K (545°F), and 705°K (810°F), respectively.

Thirdly, there must be the presence of an ignition source of sufficient energy level when the mixture temperature is below the AIT. Ignition sources include friction sparks and component surfaces heated by frictional rubbing, as well as hot chamber walls. Primary ignition sources within a sump are frictional heating of failed bearings, contact seals, and other rubbing parts.

The concept of flammability limits for lubricant vapors is important. At a given system temperature and pressure, there is an upper ratio and a lower ratio of oil vapor to air, known as the upper flammability limit (UL) and the lower flammability limit (LL), respectively, within which self-sustaining or self-propagating flames can be produced by an ignition source. At oil concentrations above the UL, the mixture is said to be too rich to burn; below the LL, it is too lean to burn (5, 6).

It is worth emphasizing that it is the oil concentration in the vapor state that defines the flammability of the oil-air mixture. The maximum concentration of oil vapor is determined by its equilibrium vapor pressure at any given temperature. The equilibrium oil-air ratio by volume is therefore the ratio of the vapor pressure of the oil to the air pressure in the chamber. The air flow rate can also affect the residence time that oil droplets, generated by the oil jet impinging on the bearing, will remain in the sump and thus the time period afforded the droplet to reach the temperature where it will evaporate or reach temperature equilibrium with the air. The air velocity can also determine if a fire will be self-sustaining. If the air velocity is greater than flame velocity the flame will be carried out of the sump with the air and thus the fire will go out when the ignition source is eliminated.

Maximum burning velocity is achieved when a stoichiometric ratio C_s of oil vapor and oxygen exist in the chamber. This ratio is equivalent to the molar ratio of oil and oxygen in the balanced chemical equation for complete combustion of the oil. The stoichiometric ratio is always within the flammability range of the oil. It has been shown for many hydrocarbons that at 297°K (75°F)

$$LL_{297^{\circ}K} (75^{\circ}K) = 0.55C_s \quad 1$$

$$UL_{297^{\circ}K} (75^{\circ}K) = 4.8C_s \quad 2$$

The flammability range increases with temperature according to the following equations:

$$LL_T = LL_{297^{\circ}K} [1 - 7.2 \times 10^{-4} (T - 297)] \quad 3$$

$$LL_T = LL_{75^{\circ}F} [1 - 4 \times 10^{-4} (T - 75)]$$

$$UL_T = UL_{297^{\circ}K} [1 + 7.2 \times 10^{-4} (T - 297)] \quad 4$$

$$UL_T = UL_{75^{\circ}F} [1 + 4 \times 10^{-4} (T - 75)]$$

By definition, oils will not burn below their flash point. Therefore, for oils with flash points higher than 297°K (75°F), the LL and UL at 297°K (75°F), calculated from equations (1) and (2), have no physical meaning but can be used in equations (3) and (4) to estimate flammability limits above the flash point. The calculated LL line should intersect the vapor pressure-temperature curve near the flash point of the oil, and this temperature, T_L , is defined as the lower flammability temperature at equilibrium vapor pressure conditions. Similarly, an upper flammability temperature T_U exists where the calculated UL line intersects the vapor pressure-temperature curve.

1.3 Importance of Engine Sump Sealing

The potential fire conditions in an aircraft engine are greatly influenced by the efficiency of the engine sump sealing system. Figure 1 is a cross-sectional view of the sump for a typical engine bearing compartment. The essential problem is to protect the bearing sump from the hot environment, which is compressor discharge air at temperatures to 922°K (1200°F) and pressures to 242 N/cm² (350 psi). (The compressor discharge air is used to cool the high-pressure-turbine disks). A buffer type of seal system is used and this requires three sets of labyrinth seals on each side of the bearing. Figure 2 is a simplified schematic of this sealing system. The buffer gas is seventh-stage compressor bleed air with a relatively low pressure of 55 N/cm² (80 psi) and temperature of 473°K (400°F); therefore, it can be allowed to leak through the inner labyrinth seal directly into the bearing compartment. This buffer gas thermally insulates the bearing compartment. The buffer system requires an overboard vent. The buffer gas flowing into this vent prevents the hotter compressor discharge air from getting into the bearing compartment. In some engines, the labyrinth seals next to the bearing compartment have been replaced with face-contact seals. This reduces leakage and results in lower specific fuel consumption. However, failure of either the labyrinth or face-contact seals could create conditions that would result in a sump fire (i.e., a rubbing friction ignition source and a hot air-oil mixture). This fact stresses the importance of developing better and more reliable seals that could reduce the probability of sump fires occurring.

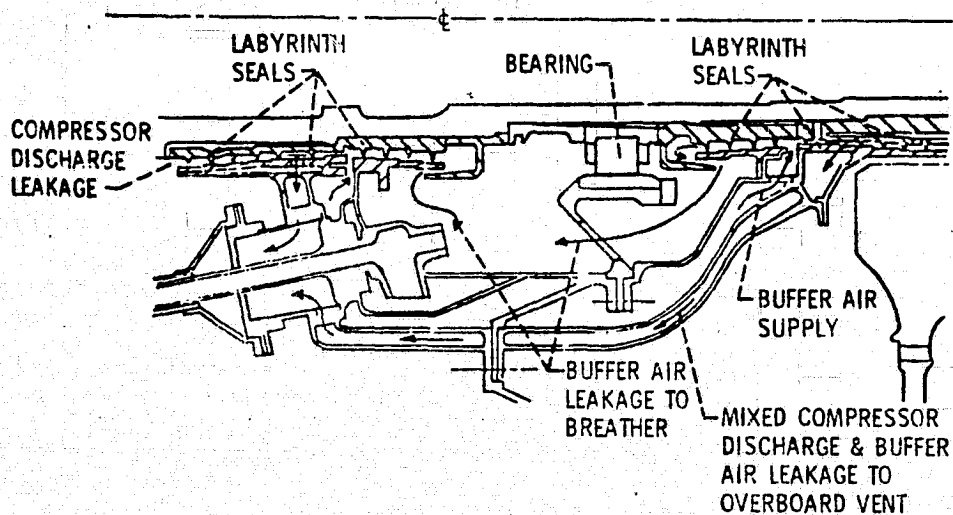


FIGURE 1

Typical Engine Bearing Sump

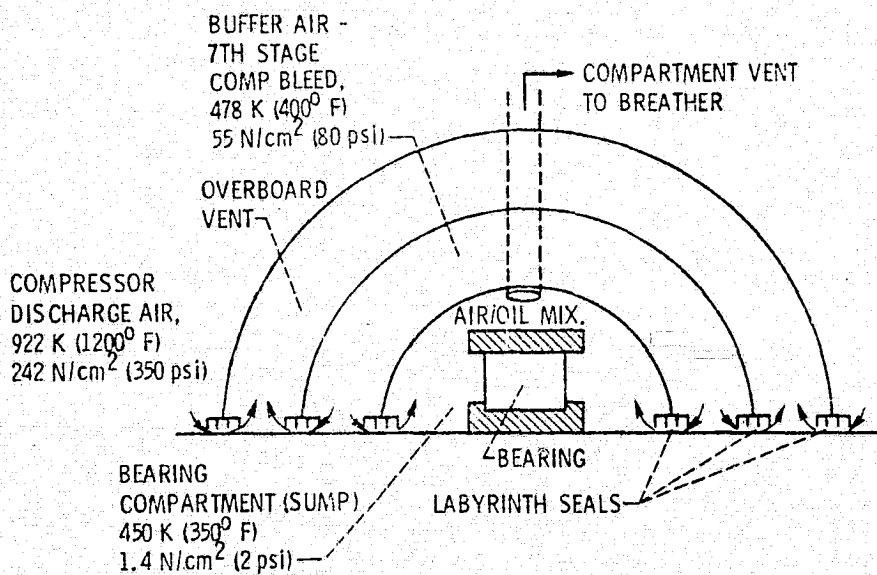


FIGURE 2

Schematic of Typical Aircraft Gas-Turbine-Engine Sump Seal System

1.4 Engine Operating Parameters Affecting Fires

The basic parameters that control fire conditions in an engine bearing sump are considered to be:

- (1) Oil flow rate into the sump
- (2) Oil inlet temperature
- (3) Air leakage rate to the sump
- (4) Air inlet temperature
- (5) Shaft or bearing speed
- (6) Ignition source and duration

Other parameters, such as sump volume and geometric configuration as well as bearing, shaft, seal, and housing temperatures and lubricant flammability, can also affect sump-fire susceptibility.

2.0 TEST FACILITY

All tests were performed in an existing NASA owned high speed bearing test rig modified to accomodate the requirements for the fire study. The rig operates with an SKF owned drive, oil and air systems and controls. The basic test equipment is shown diagrammatically in Figure 3 and consists of the following components:

- Test Rig
- Drive System
- Hot Air System
- Lubrication System
- Nitrogen Purging System
- Instrumentation
- Fire Ignition System
- Oil Recovery System

2.1 Test Rig

The basic test rig is designed to simulate aircraft engine mainshaft designs by avoiding thick sections in the shaft and bearing housings and by introducing flexible sections between the main rig outer housing and the bearing outer rings. This flexibility is intended to simulate to some extent the self-aligning ability of current aircraft engine bearing mounts. A drawing of the cross-section of the test bearing and sump area of the test rig is presented in Figure 4.

The test rig consists of a 0.3 meter (12 inch) diameter cylindrical housing in which a hollow shaft of approximately 0.13 meter (5 inch) maximum diameter is supported by the test bearing at one end and a cylindrical roller (rig) bearing at the other. The housing itself is mounted in a horizontal position above a table by means of a special support system which maintains the center line height and parallelism with the table while freely permitting both radial and axial thermal expansion. This arrangement is best shown by the isometric sketch in Figure 5.

The pedestal is positioned in the plane of the test bearing for optimum rigidity and the sliding ring in the plane of the roller bearing. The pedestals are bolted securely to the rig table and also dowelled to maintain alignment.

FIGURE 3
Schematic of Test Rig Layout

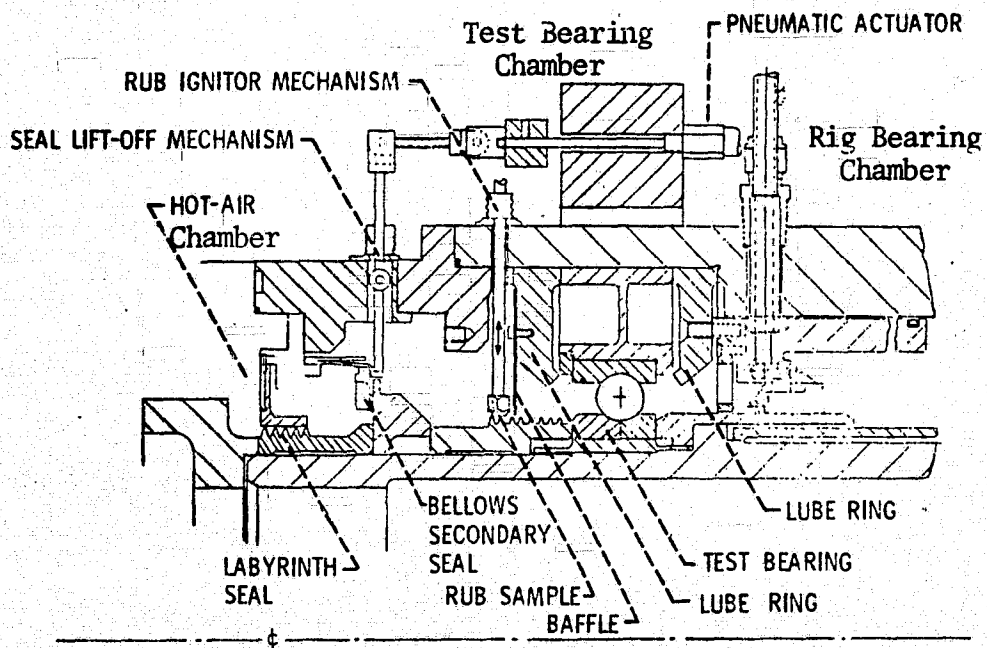


FIGURE 4
Sump Fire Test Rig

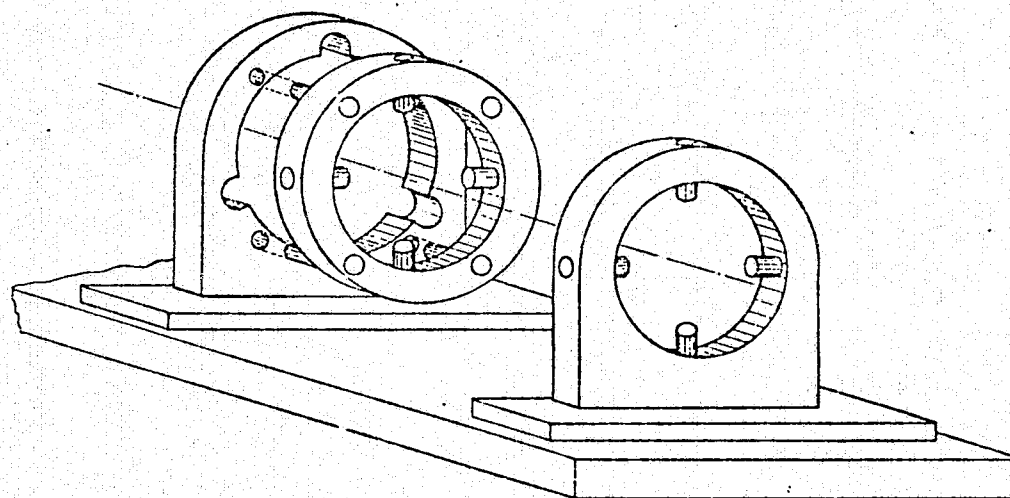


FIGURE 5

Isometric View of Mounting Arrangement

The inside of the rig is divided into three basic compartments; 1) Hot air chamber 2) Test bearing chamber 3) Rig bearing chamber, see Figure 4.

The hot air chamber is that space forward of the bellows seal and is maintained at a pressure necessary to supply the thrust load to the test bearing and hot air flow into the test bearing chamber during lift-off of the bellows seal for fire ignition attempts. Three pneumatically actuated rams which provide lift-off or opening of the bellows seal are mounted circumferentially at 180° increments on the forward pedestal and the actuation linkages enter the hot air cavity through flexible metal seals, see Figure 6.

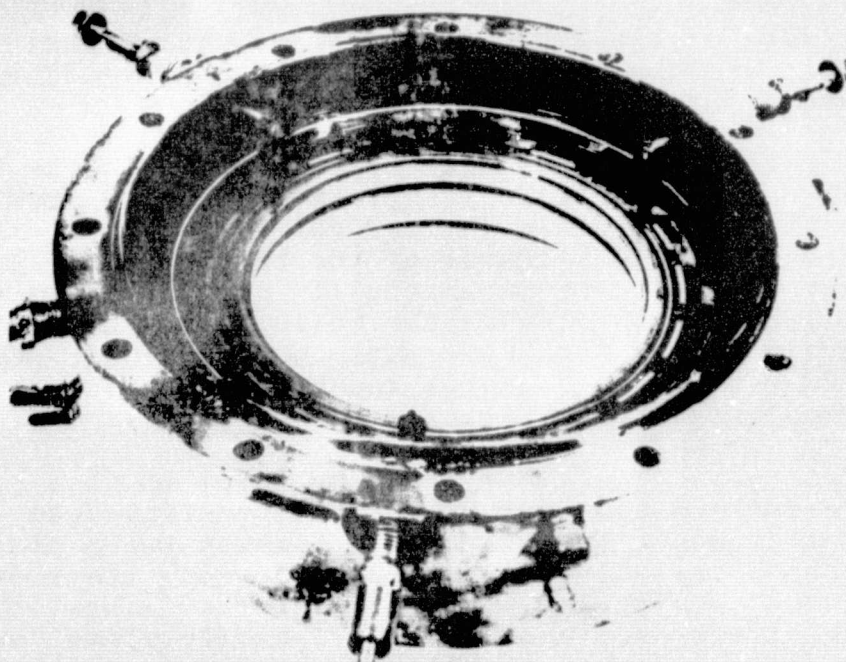
The test bearing chamber (bearing sump) is that space between the bellows seal and the circumferential seal and provides the environment in which fire ignition takes place. This chamber accomodates a 125 mm bore ball bearing (test bearing) mounted in a I housing and two lubricant feed rings, one on either side of the bearing. A hot air baffle plate, see Figure 6, used to retard the hot air flow directly on to the bearing during bellows seal lift-off, is mounted outboard of the lubricant ring located forward of the bearing. Two oil drain holes are located at the base of the chamber housing, one on either side of the bearing and a hot air exhaust port approximately 45° from the top on the outboard side of the baffle plate.

The rig bearing chamber, that space between the circumferential and rig seals, contains the roller (rig) bearing which supports the back end of the shaft, the lubricating ring for the rig bearing, the rig bearing housing, and the lubricant drain port.

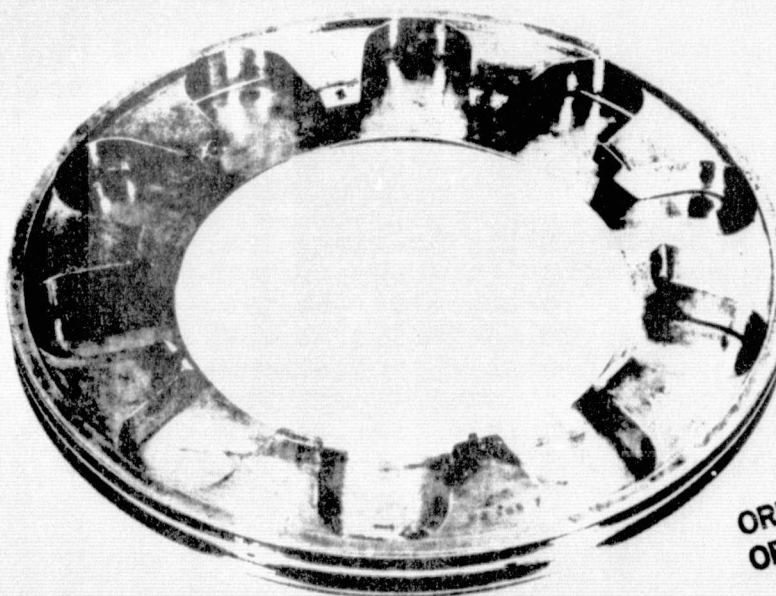
Both the ball and roller bearings used to support the rotating shaft are mounted in I shaped housings to provide sufficient flexibility to minimize shaft-to-housing misalignment forces produced by uneven thermal gradients. The bearings are mounted to the shaft through a specially designed sleeve which compensates for unequal thermal growth between the shaft and bearing bore. This prevents excessive mounting stresses and deflections which would affect the internal clearance in the bearings.

FIGURE 6

RIG COMPONENTS



Seal and Seal Housing Showing Lift-Off Mechanism



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Lubrication Jet Ring and Baffle Plate

2.2 Drive System

The test rig shaft is driven by a motor and jackshaft assembly. The variable speed, 50 HP DC drive motor is mounted on an adjustable base and drives the jackshaft through a flat belt. The jackshaft unit consists of a hollow shaft mounted in matched pairs of preloaded angular contact bearings at each end with a 0.076 meter (3 inch) diameter removable pulley at the center. The bearings are supported in steel pillow blocks bolted to a rigid base, and are lubricated by a circulating cold mineral oil supply fed to the top cap of each bearing.

The rig shaft is connected to the jackshaft by a flexible coupling. The other end of the jackshaft drives a tachometer through a small flexible coupling. The jackshaft like the rig shaft is dynamically balanced for high speed operation. A schematic of the drive system is presented in Figure 7.

2.3 Hot Air System

The air flow commences with an air compressor which has a rated output of 2.57 scmm (91 scfm) at 1.4×10 newtons/sq. meter (200 psig). Air feeds directly to a dryer and filter column which reduces the moisture content to a 228°K (-50°F) dew point and the hydrocarbons to 13 parts per million. This clean, dry air then passes to a 0.566 cubic meters (20 cu. ft.) receiver and hence through a shut-off valve and to a pressure regulator. A pneumatic servo control on this regulator maintains the desired pressure in the rig air chamber. The regulated air then passes through a 45 kw electrical heater in which the air passes through approximately 6.7 meters (22 ft.) of 316 stainless steel tubing which is radiantly heated by the electric elements, and hence to the rig hot air chamber.

The output of a thermocouple mounted in the hot air line between the heater and rig is fed to the electrical input controls of the 45 kw heater so the desired temperature can be maintained.

Air entering the hot air chamber of the rig is either exhausted through a pneumatically actuated hot air chamber exhaust valve and/or through the bellows seal into the bearing chamber where it exhausts through a hot air port. The air passing through the test bearing chamber passes through a flow meter before being exhausted external of the test cell through a stack.

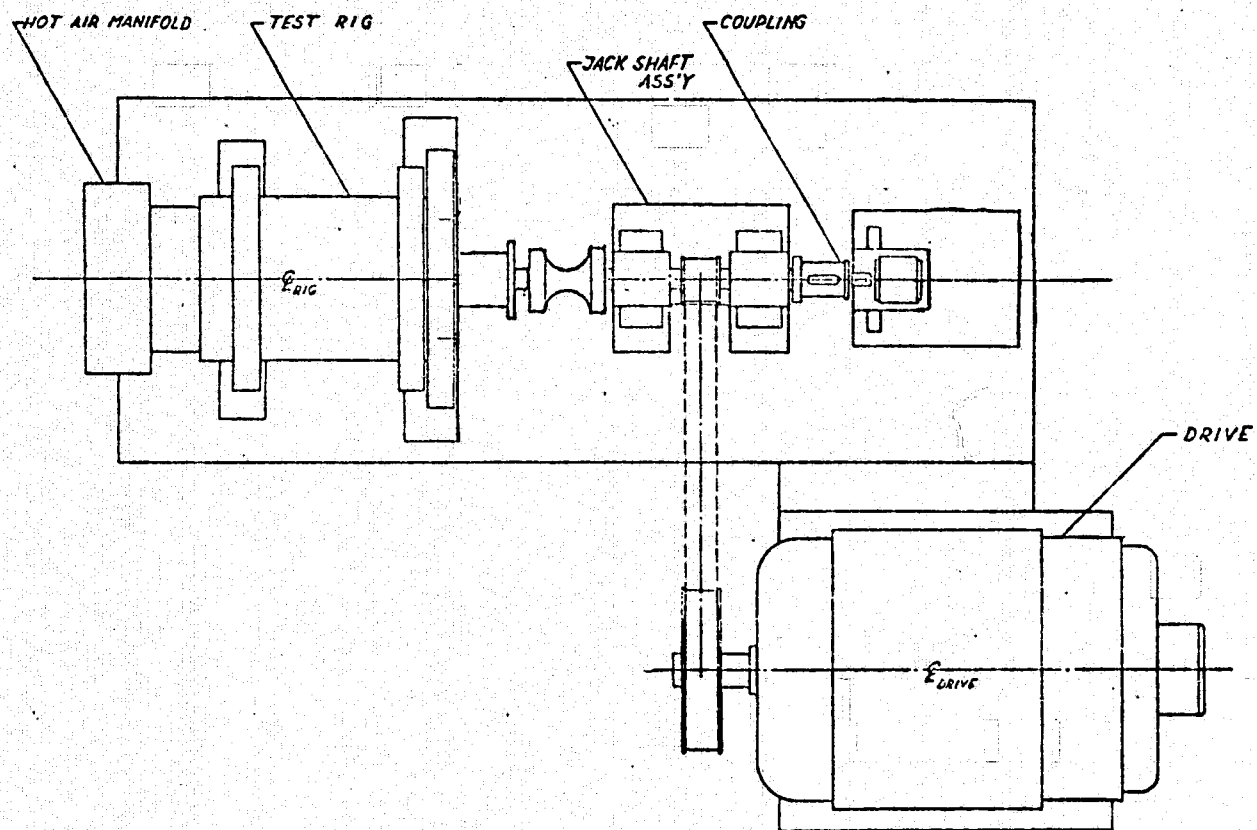


FIGURE 7

Test Rig Drive System

2.4 Lubrication Systems

The 125 mm bore ball bearing (test bearing) and the roller bearing (rig bearing) are lubricated by separate lubrication systems. This arrangement was implemented to minimize the safety hazard by using a non-flammable fluid (Drytox) for lubricating the rig bearing and thus preventing the propagation of fires from the test bearing chamber into the rig bearing chamber. The test bearing lubrication system was purchased from AR Dervares Company in accordance with SKF specifications and consists of the following components.

- Oil Storage and Heating Tank
- Filter Unit
- Lubrication Pump
- Bypass Valve
- Scavenge Pump

The test oil is stored and heated by electrical resistance heaters in a 0.057 cu m (15 gallon) capacity, thermally insulated tank with a level indicator and oil temperature sensor. The oil is pumped from the storage tank through stainless steel tubing into a Filterite, six element 10 micron (absolute) filtering unit by a Viking model 64724 gear pump driven by a 1/2 HP constant speed AC motor. The oil then passes through a Cashco model 460 bypass valve allowing excess oil to return to the storage tank. The oil flowing to the test rig then passes through a pneumatically actuated, variable orifice, flow control valve and then through a Brooks model 103623w-5510A flow meter. The oil leaving the flow meter is divided equally into two paths each entering one of the two oil manifold rings inside the rig. Each oil manifold ring contains ten equally spaced nozzles, see Figure 6, (four nozzles plugged to increase jet velocity) which direct the oil flow against the bearing sides. A Viking gear pump, model HL 4724, driven through a variable speed drive by a 3/4 HP AC motor scavenges the oil from the bearing chamber.

The lubrication system provided for the rig bearing is similar in design to the test bearing system with the exception that different components are used and no scavenge pump is incorporated in the return line (gravity feed only).

2.5 Nitrogen Purging System

The need to provide an inert gas for purging fires in the test rig was accomplished by piping a nitrogen supply line directly into the test bearing chamber. The nitrogen is supplied from five standard nitrogen tanks connected to provide a large supply of nitrogen gas. A spring loaded quick release valve in the line permits rapid injection of the gas into the bearing chamber.

2.6 Instrumentation

The test rig was instrumented to measure the following parameters:

- Hot Air Chamber Pressure
- Hot Air Inlet Flow Rate
- Test Bearing Oil Inlet Flow Rate
- Rig Bearing Oil Inlet Flow Rate
- Test Bearing Oil Scavenge Rate
- Test Bearing Oil Inlet Temperature
- Hot Air Inlet Temperature
- Bearing Outer Ring Temperature
- Shaft Speed
- Oil-Air Temperature in Test Bearing Chamber

A Honeywell Model Y702X21-C39-II-III dual pen pressure recorder was used to measure the pressure in the hot air chambers. The recorder is located outside the test cell and is actuated through 0.635 cm (1/4 inch) copper tubes connected to the proper rig chambers. In addition to the recorder, two pressure transducers send electrical signals proportional to pressure to meters located on the instrument console.

The hot air inlet flow rate was measured by a Brooks Rotameter model 1140 which is located in the exhaust line from the sump.

The test bearing and rig bearing lubrication flow rates were measured just prior to the lubricants entering the rig. The test and rig bearing lubricant flow rates were measured by a Brooks Armored Rotameter model 103623w-5510A and a Brooks Rotameter model 1110-0903PB1A respectively.

The oil scavenge flow rate from the test bearing chamber is fixed by the shaft speed of the Viking gear pump model HL4724. A calibration of the shaft speed with respect to the variable speed drive control was performed prior to testing and thereafter the scavenge rate was established by the setting on the variable speed drive.

All required temperature measurements were made with iron-constantan thermocouples except the seven thermocouples located in the sump which were chromel-alumel. The chromel-alumel thermocouples were connected directly to Honeywell strip chart recorders. All the other thermocouples were attached to a patch panel, outside the test cell, from which they were connected to a Honeywell strip chart recorder for a continuous record or an Esterline Angus multipoint recorder which records each point approximately once every 72 seconds. During fire study testing, the hot air inlet temperature and the hot air exhausting from the bearing chamber were recorded on the strip chart while all other temperatures were recorded on the multipoint recorder.

The shaft speed was controlled manually by a variable-speed drive and the speed monitored by a tachometer-generator mechanically coupled to the test shaft and producing electrical impulses proportional in number to the rotational speed of the shaft. The output signal was presented on a Hewlett Packard electronic counter model 521CR in cycles per minute.

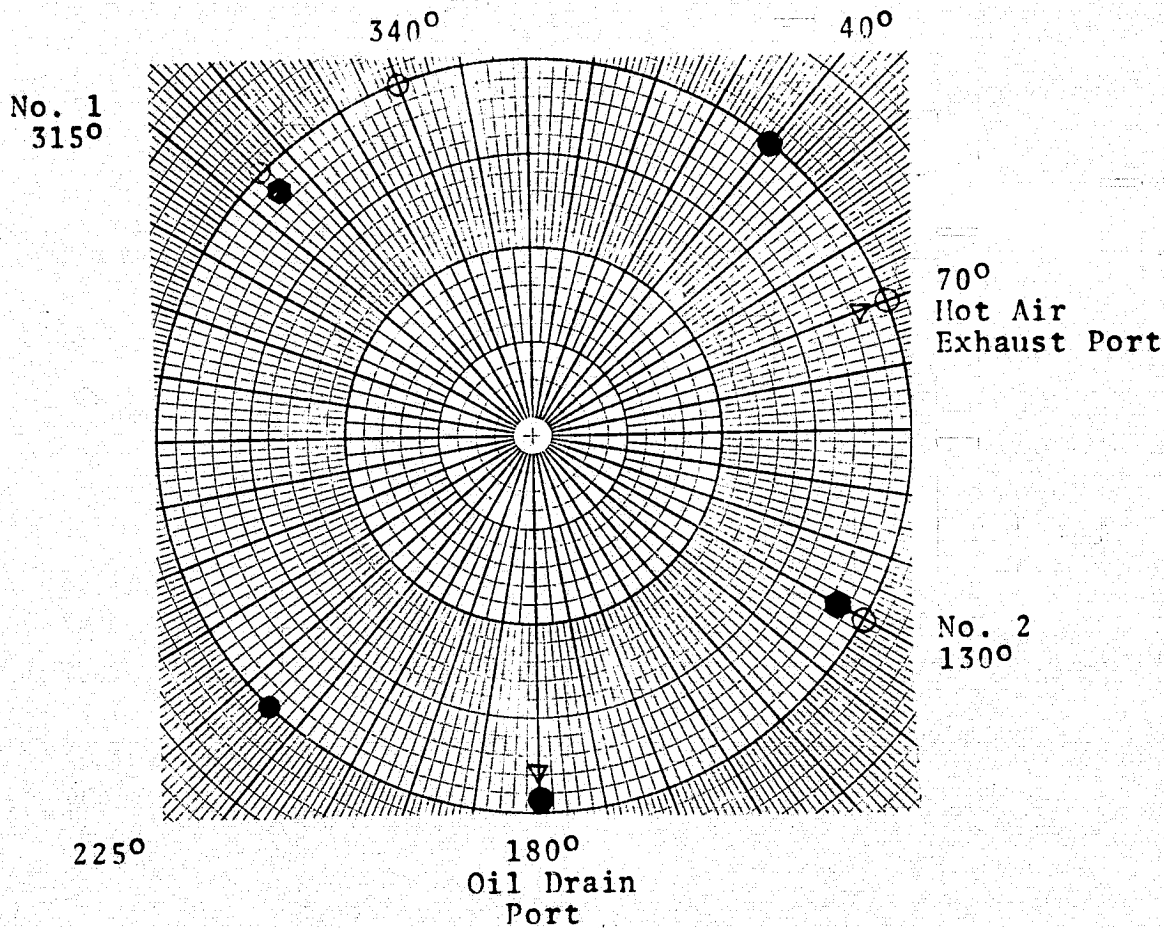
2.7 Fire Ignition System

Two different types of fire ignition devices were incorporated into the test rig to aid in the generation of fires. The primary system installed was a spark generator actuated by a 10,000 volts, 23 ma AC signal produced by a Jefferson Electric Company Ignition transformer operating from a 120 volt 60 cycle input. Two spark generators were installed through the rig housing into the forward section of the sump. The ignitor tips were positioned to produce sparking to the hot air baffle plate at approximately 4 and 10 o'clock. An ammeter was installed into the primary circuit of the transformer to detect sparking. The location of the spark ignitors, sump thermocouples, and hot air exhaust port are shown in Figure 8.

FIGURE 8

LOCATION OF THERMOCOUPLES, EXHAUST PORTS AND SPARK IGNITORS

- △ Exhaust Ports Locations
- Spark Ignitor Locations
- Location of Thermocouples
In Plane 0.64cm in Front of Baffle Plate
- Location of Thermocouples
In Plane 3.8cm in Front of Baffle Plate



2.8 Oil Recovery System

The high rate of test oil loss, carried by the hot air exhausting from the bearing chamber, during the testing dictated the use of an oil recovery system. The system incorporated a large .91x.61m (3 x 2 ft) separator tank assembled onto the bearing chamber hot air exhaust line and a return line attached directly to the test bearing lubrication supply tank. The large size of the tank and a baffle plate located between the inlet and exhaust parts converts the high velocity of the gas carrying the oil to a static pressure allowing the oil to migrate to the bottom of the tank. The separated oil accumulated in the bottom of the tank was returned to the supply tank by a positive displacement pump located on top of the tank. The hot air flowing through the tank prevents excessive cooling of the oil.

3.0 TEST MATERIAL

All tests on this program were performed with a lubricant having an ester-base formulation meeting the MIL-L-23699 specifications. In its present formulation more stable base stock components have been used which were not previously available. This lubricant was chosen because of its wide usage in gas turbine engines for bearing lubrication. The lubricant has a flash point temperature of 525 K° (485°F), a fire point temperature of 558 K° (545°F), and an autoignition temperature (AIT) of 705 K° (810°F). The following other properties were either acquired from the vendor or approximated by calculations based on a singular molecular structure.

Molecular Weight - gms/mole	556
Vapor Pressure - N/m ² (pisa)	46.2 at 490°K (0.0067 at 425°F) 137.9 at 518°K (0.02 at 475°F)
Specific Heat of Liquid Fuel-J/kg°K (Btu/lb _m °F)	2246 at 310°K (0.536 at 100°F) 2351 at 421°K (0.561 at 300°F) 1886 at 505°K (0.450 at 450°F)
Specific Heat of Gaseous Fuel-J/Kg°K (Btu/lb _m °F)	1257 (0.3)
Enthalpy of Formation of Liquid Fuel J/°K (Btu/lb _m)	118.8 x 10 ⁵ (6250)
Enthalpy of Formation of Gaseous Fuel - J/°K (Btu/lb _m)	120.3 x 10 ⁵ at 422°K (6330 at 300°F) 121. x 1 x 10 ⁵ at 478°K (6375 at 400°F)

4.0 TESTING PROCEDURE

Testing was initiated after obtaining the desired inlet parameter conditions, shaft speed, and the temperatures had remained stable in the bearing chamber for approximately ten minutes. With the desired air and oil flow into the bearing chamber, the spark ignitor (the ignition source) was then activated until a fire was ignited or for a maximum period of 60 seconds if a fire was not ignited. If the fire was not self-extinguishing when the ignitor was turned off, the nitrogen purging system was activated and the bearing chamber purged with nitrogen gas. In either case, fire or no fire ignited, the next fire ignition attempt was performed either with the second ignitor or after one or more of the input variables were changed and stabilized temperatures obtained. The specific test parameter or combination of test parameters changed between each ignition attempt varied for each individual test.

The parameter ranges investigated during the testing were:

Test oil inlet temperature	353-467° K (175-380° F)
Test oil flow rate	0.23-0.57 m ³ /hr (1-2.5 gpm)
Hot air temperature entering hot air chamber -	622-833° K (660-1040° F)
Hot air flow rate through bearing chamber -	7-48 std m ³ /hr (4-28 scfm)
Test bearing outer ring temperature	408-528° K (275-490° F)
Shaft rotational speed	14,000 rpm

The following procedure was followed to bring the test rig to test conditions.

1. Air compressor started and pressure adjusted to 131-138 N/cm² (190-200 psi).

2. Temperature controller for test bearing and rig bearing storage tanks set at 533° K (500° F) and 339° K (150° F) respectively. Circulation started of test bearing oil through filter, pump and bypass valve. Immersion heaters in test bearing storage tank activated and rheostats adjusted to obtain a current of 12 amps in each heater. Rig bearing lubricant heaters are automatically actuated when controller is set.

3. When test oil temperature reaches 410°K (280°F), hot air flow initiated through the test bearing chamber. Air temperature controller set at 533°K (500°F) and flow rate adjusted to 34 to 51 stdm^3/hr (20 to 30 scfm) by regulating hot air manifold pressure. The hot air flow aids in warming the rig components and the oil-air separator tank to minimize oil cooling when circulation through the rig is initiated.

4. Circulation of test oil and rig bearing oil started through rig when test oil reaches desired temperature. Test oil flow rate adjusted to desired flow rate.

5. Jack shaft lubrication system turned on.

6. Shaft accelerated slowly to a speed of 14,000 rpm.

7. Hot air temperature set at desired level.

8. Operation continued until test conditions obtained.

5.0 TEST RESULTS AND DISCUSSION

A total of four test series were conducted. Each individual test run was performed to evaluate the possibility of igniting fires within a given range of input parameters and a given sump configuration. Test Series 1 was performed with the initial baffle plate configuration which consisted of a 0.76 mm (0.030 inch) thick monel sheet attached to the outboard lubrication ring. The baffle separated the outboard section of the bearing sump into two sections with a 3.2 mm (.125 inch) radial gap around the shaft and 645 sq. mm (1 sq. in.) hole located on the outer edge at 6 o'clock providing the only flow paths between the two sections. The bearing lubricant entered the sump on the inboard side of the baffle and the hot air entered and was exhausted on the outboard side of the baffle. The test series consisted of five test runs with a total of 60 fire ignitions attempted.

Test Series 2 was performed with a modified baffle which eliminated the 645 sq. mm (1 sq. in.) opening and decreased the circumferential gap to a 0.35 mm (0.014 in.) width. In addition, an oil flinger was located on the shaft inboard of the baffle to fling the oil away from the gap and thus further decrease the amount of oil mixing with the hot air entering the sump. Two test runs were performed with 49 attempts made to ignite fires under input parameter conditions similar to those used in test Series 1.

Test Series 3 and 4 were both performed with the same baffle configuration as Series 2 with each test run starting with relatively low mixture temperatures in the sump and then more severe conditions imposed (higher mixture temperature) by increasing the inlet oil and/or air temperatures. Test Series 3 and 4 consisted of three runs each with a total of 79 fire ignition attempts in Series 3 and 52 in Series 4.

This section is presented in the chronological order of the testing. The considerations for performing each test, the methods used to interpret the results, and the conclusions reached also follow in the order of testing.

5.1 Test Series 1

The purpose of test Series 1 was to evaluate the possibility of igniting fires over a wide range of input parameter values (oil and air temperatures and flow rates) and determine if

various combinations could be detected which produced conditions in the sump which were conducive to fires being started by the electric spark ignitor.

Series 1, Run 1

The first run was performed with a simulated compressor air leakage rate into the bearing sump of 18-20 stdm³/hr. (11-12 scfm) which is considered to be representative of leakage through a labyrinth seal. This produced approximately 220 to 240 air changes per minute in the bearing sump. The extreme values of the input variables and resulting bearing and mixture temperatures were:

Hot air flow rate through sump	18.7-20.4 stdm ³ /hr (11-12 scfm)
Hot air temperature entering hot air chamber	725-745 K (843-880 F)
Oil inlet temperature	438-45 K (330-350 F)
Oil flow rate	0.23-0.46 m ³ /hr. (1-2 gpm)
Bearing outer ring temperature	443-494 K (339-430 F)
Temperature at ignitor (mixture temperature)	522-539 K (480-510 F)

In 10 of the 16 activations of the spark ignitor, fires were started. However, the propagation of the fires was relatively slow and none were self-sustaining. Conditions around spark ignitor number 2 (4 o'clock) were more susceptible to fire ignition over the range of conditions tested as evidenced by only two fires out of eight attempts being started at ignitor 1 while fires were ignited in each of the eight attempts using ignitor 2. The two fires that were started by ignitor 1 occurred when the most severe conditions (highest oil and air inlet temperatures) were imposed.

Series 1, Run 2

The second run was performed to evaluate the possibility of igniting fires when the simulated air leakage rate and the air temperature entering the hot air chamber were increased. The air leakage rate was increased first to 40.8 stdm³/hr. (24 scfm) and then 47.6 stdm³/hr. (28 scfm) which are con-

sidered to be representative of air flow rates through either a damaged labyrinth or face seal. The extreme values of the test variables were:

Hot air flow rate through sump	40.8-47.6 stdm ³ /hr (24-28 scfm)
Hot air temperature entering hot air chamber	786-833°K (955-1040°F)
Oil inlet temperature	433-444°K (320-340°F)
Oil flow rate	0.23-0.46 m ³ /hr (1-2 gpm)
Bearing outer ring temperature	478-520°K (400-475°F)
Temperature at ignitor (mixture temperature)	588-677°K (600-760°F)

A total of 20 attempts were made to ignite fires. In 11 attempts, fires were ignited, of which 8 were self-sustaining. With the lower of the two air flow rates 40.8 stdm³/hr. (24 scfm) and inlet temperature 876°K (955°F) no fires could be started with ignitor 1 at any of the three oil flow rates 0.23, 0.34, and 0.46 m³/hr. (1, 1.5 and 2 gpm). Using ignitor 2, fires were started at all oil flow rates; however, those started at flow rates of 0.23 and 0.34 m³/hr. (1 and 1.5 gpm) were not self-sustaining. The fire ignited at an oil flow rate of 0.46 m³/hr. (2 gpm) was self-sustaining. With the higher air flow rate 47.6 stdm³/hr. (29 scfm) and temperature 833°K (1040°F) self-sustaining fires were obtained with ignitor 1 when oil flow rates of 0.34 and 0.46 m³/hr. (1.5 and 3 gpm) were supplied. No fire was started in two attempts when only 0.23 m³/hr (1 gpm) of oil was supplied. Using spark ignitor 2, self-sustaining fires were started at all three supply rates.

In general, the results of run 2 indicated, even though the percentage of fires ignited was less, that the combined higher air flow rates and temperatures resulted in conditions susceptible to self-sustaining fires. The test data also indicated, that under the more severe conditions, the quantity of oil supplied to lubricate and cool the bearing had a definite effect on fire ignition. The most susceptible condition for fire ignition resulting from the higher oil flow rate.

Series 1, Run 3

The purpose of the third run was to further evaluate the ignition of fires at the high air flow rate, but with a lower air inlet temperature. In addition, the effect of shaft or bearing speed on the conditions within the sump were checked. A total of 9 attempts were made to ignite fires using only spark ignitor 2. The extreme values of the test variables were:

Hot air flow rate through sump -	47.6 stdm ³ /hr. (28 scfm)
Hot air temperature entering hot air chamber -	732-788 ^o K (860-960 ^o F)
Oil inlet temperature -	421-439 ^o K (300-330 ^o F)
Oil flow rate -	0.23-0.46 m ³ /hr. (1-2 gpm)
Bearing outer ring temperature -	455-500 ^o K (360-440 ^o F)
Temperature at ignitor (mixture temperature) -	561-622 ^o K (550-660 ^o F)

With all conditions held essentially constant, except for the bearing temperature which increased from 732 to 743^oK (360-380^oF) due to the speed increases, fire ignitions were attempted at shaft speeds of 7,000, 10,500, and 14,000 rpm. At 7,000 rpm, no fire could be ignited. At 10,500 rpm a temperature increase and decrease of approximately 56^oK (100^oF) was noted three times during the 60 second spark ignition actuation, indicating that a mild fire was being ignited, but could not sustain itself even with the ignition source retained. At 14,000 rpm, a self-sustaining fire was ignited. During this incremental speed increase period, the mixture temperature at the spark ignitor decreased from 622^oK (660^oF) to 594^oK (610^oF) and then to 572^oK (570^oF) even though the oil flow rate was maintained at 0.46 m³/hr. (2 gpm) and the air inlet temperature was held constant. This condition indicated that the increased speed resulted in a greater dispersion of oil and thus more heat removed from the air as oil vapor was formed. It is also feasible that the lower measured temperatures were due partially to more oil droplets impinging on the thermocouple.

Following the speed effect evaluation, the air inlet temperature was reduced from 786°K (955°F) to 730°K (870°F) while maintaining the high air flow rate through the sump. A self-sustaining fire was readily ignited with an oil flow rate of $0.46 \text{ m}^3/\text{hr.}$ (2 gpm). At an oil flow rate of $0.34 \text{ m}^3/\text{hr.}$ (1.5 gpm) fires were ignited in both attempts; however, they were not self-sustaining. With $0.23 \text{ m}^3/\text{hr.}$ (1 gpm) of oil supplied, fires were not ignited in two out of three attempts. As the oil flow rate was reduced in the steps listed above, the mixture temperature increased first from 560°K (550°F) to 588°K (600°F) and then to 617°K (650°F). This increase in temperature indicated that less oil was mixing with the air as the oil flow rate was decreased.

The results of this run showed that a drop in the hot air inlet temperature of approximately 56°K (100°F) had little or no effect on the susceptibility to fire ignition when the air flow rate was high. However, the variation in the oil flow rate, as well as the increased dispersion due to increased bearing speed substantiated the evidence observed in Run #2 which suggested that the mixture was too lean to burn when low oil flow rates were supplied. It is also feasible that at the higher oil flow rates and higher bearing speeds the oil dispersion was not only greater, but the size of the droplets formed were smaller thus permitting a greater transfer of heat to the oil and thus greater evaporation.

Series 1, Run 4

The purpose of Run #4 was to evaluate the influence of lower inlet oil temperatures on fire ignition. The test run was performed with a high air temperature entering the hot air chamber and initiated with a low oil temperature which was increased in incremental steps between fire ignition attempts. The extreme values of the test variables were:

Hot air flow rate through sump -	37.4-44.2 stdm ³ /hr. (22-26 scfm)
Hot air temperature entering hot air chamber -	800-816 ^o K (980-1010 ^o F)
Oil inlet temperature -	352-418 ^o K (175-295 ^o F)
Oil flow rate -	0.34-0.46 m ³ /hr. (1.5-2 gpm)
Bearing outer ring temperature -	416-468 ^o K (290-385 ^o F)
Temperature at ignitor (mixture temperature) -	482-583 ^o K (410-590 ^o F)

A total of eight attempts were made to ignite fires using ignitor 2 in all cases. In the first four attempts, during which the oil inlet temperature was increased from 352°K (175°F) to 400°K (260°F) in approximately 15°K (27°F) increments, no fires could be ignited. In the next four attempts, during which the oil inlet temperature was maintained at approximately 419°K (293°F) non self-sustaining fires were ignited in all cases even though the oil inlet flow rate was decreased from 0.46-0.34 m³/hr. (2-1.5 gpm) for the final two ignitor actuations. However, the non self-sustaining fires were less severe (low temperature increase) with the lower flow rate and the mixture temperature increased 22°K (40°F). The test data also showed that while maintaining the air inlet temperature to the hot air chamber at approximately 810°K (1000°F) the mixture temperature at the ignitor increased 39°K (70°F) while the oil inlet temperature was increased 36°K (65°F); thus showing the effect of a change in oil temperature.

The results of run 4 showed that a low oil inlet temperature prevented conditions within the sump from reaching a state where fires could be ignited. The results also showed that the oil inlet temperature had an appreciable influence on the mixture temperature and once again indicated that decreasing the oil flow rate resulted in conditions less susceptible to fires.

Series 1, Run 5

Test run 5 was performed to evaluate if conditions within the sump were less susceptible to fire ignition when the air flow rate was representative of that which would occur when a properly operating face seal was used between the compressor discharge air and the oil sump. The extreme values of the test variables used in this run were:

Hot air flow rate through sump -	6.8-13.6 stdm ³ /hr. (4-8 scfm)
Hot air temperature entering hot air chamber -	744-805°K (880-990°F)
Oil inlet temperature -	438-460°K (330-370°F)
Oil flow rate -	0.23-0.46 m ³ /hr. (1-2 gpm)
Bearing outer ring temperature -	489-510°K (420-460°F)
Temperature at ignitor (mixture temperature) -	522-545°K (480-520°F)

A total of seven attempts were made to ignite fires using ignitor 2 in all cases. Two ignition attempts were made with oil flow rates of 0.34 and 0.46 m³/hr. (1.5 and 2 gpm) and three attempts with 0.23 m³/hr. (1 gpm). In all cases very mild fires were ignited, evidenced by 6-110°K (10 to 200°F) temperature increases in the sump, which were not self-sustaining even with the ignitor continuously activated.

The results of this run indicated that conditions within the sump were not as susceptible to fire ignition when the air leakage rate was maintained at a low value or at a value which would be expected when a properly operating face seal is used.

Test Series 1 Summary

In order to obtain a better physical picture of what was occurring within the sump, the temperatures measured at each of the thermocouple locations, see Figure 8, just prior to and following an attempted fire ignition were tabulated for each run, see Tables 1-5. In addition, plots were made of the temperature data before and after a fire ignition (either self-sustaining or non self-sustaining) for one attempted ignition at each of the three oil flow rates for the first three test runs, see Figures 9-12.

The most obvious information observed from the tables is the drop in the hot air temperature while passing through the hot air chamber and the bearing sump before reaching a thermocouple location. This decrease in temperature is obviously the result of heat transfer to the surrounding metal and the oil. It is also observed that the mixture temperature at each thermocouple location within the bearing sump differs, with the temperatures at the air and oil exhaust ports (azimuth location of 70° and 180° respectively) being generally lower. This is the result of a greater quantity of heat being transferred from the air to the oil making the mixture temperature lower just before it exhausts from the sump (longer residence time of oil drops in air).

TABLE 1
AIR TEMPERATURE IN BEARING SUMP
SERIES 1 RUN 1

Shaft Speed (rpm)	Attempted Fire Ignition	Igniter Activated	Result	Oil Inlet Temp. (°F)	Oil Flow (gpm)	Hot Air Inlet Temp. (°F)	Air Flow Rate (scfm)	Mixture Temp. (°F) at Each Thermocouple Location											
								40°	40°+	70°	70°+	130°	130°+	180°	180°+	225°	225°+	340°	340°+
14,000	1	1	*NSSF	330	1	880	11	560	650	500	660	500	600	500	640	630	700	515	630
14,000	2	1	NSSF	330	1	880	11	560	790	500	1280	500	1090	500	1270	620	820	520	940
14,000	3	1	NF	330	1.5	870	11	540	540	500	500	520	520	480	480	590	590	500	500
14,000	4	1	NF	330	1.5	870	11	530	530	500	500	520	520	490	490	590	590	500	500
14,000	5	2	NSSF	330	1.5	870	12	540	640	500	800	510	710	480	700	590	690	500	680
14,000	6	2	NSSF	330	1.5	870	12	530	670	480	740	510	730	480	950	580	745	500	700
14,000	7	2	NSSF	320	2	870	12	530	660	480	860	500	680	460	670	570	665	480	690
14,000	8	2	NSSF	320	2	870	12	510	690	490	930	480	720	470	900	580	675	480	690
14,000	9	1	NF	320	2	865	12	510	585	485	400	480	505	470	470	580	580	480	485
14,000	10	1	NF	320	1	858	12	535	540	470	470	495	500	480	485	590	590	495	500
14,000	11	1	NF	320	1	858	12	530	540	480	475	500	480	490	495	590	590	500	500
14,000	12	2	NSSF	320	1	858	12	540	670	480	1020	500	800	480	1020	500	710	500	750
14,000	13	2	NSSF	350	1	845	12	545	640	480	1000	500	795	495	1000	590	720	525	745
14,000	14	2	NSSF	350	1	880	12	540	600	460	750	500	670	470	750	580	655	500	660
14,000	15	2	NSSF	345	1	880	12	540	630	470	940	510	760	480	1000	580	580	500	740
14,000	16	1	NF	340	1	880	12	530	540	460	460	500	510	480	480	580	580	500	500

*NF - No Fire
 NSSF - Non Self-Sustaining Fire
 SSF - Self-Sustaining Fire
 + - Temperature After Attempted
 Fire Ignition

TABLE 2
TEST DATA
SERIES 1 RUN 2

Shaft Speed (rpm)	Attempted Fire Ignition	Ignitor Activated	Result	Oil Inlet Temp. (°F)	Oil Flow (gpm)	Hot Air Inlet Temp. (°F)	Air Flow Rate (scfm)	Mixture Temp. (°F) at Each Thermocouple Location													
								40°	40°+	70°	70°+	130°	130°+	180°	180°+	225°	225°+	315°	315°+	340°	340°+
14,000	1	1	*NF	325	1	955	24	680	680	530	530	630	640	600	610	695	695	680	680	620	630
14,000	2	1	NF	320	1	955	24	680	680	520	520	620	620	600	610	695	695	680	680	620	630
14,000	3	2	NF	335	1	960	24	680	680	540	650	640	680	600	650	700	700	680	690	620	630
14,000	4	2	NSSF	325	1	960	24	680	710	530	1300	640	750	550	790	700	700	620	720	620	750
14,000	5	2	NSSF	330	1.5	960	24	620	710	520	1270	600	740	590	820	690	690	680	720	500	720
14,000	6	2	NSSF	340	1.5	960	24	600	730	530	1290	600	780	570	900	690	690	680	760	560	730
14,000	7	1	NF	330	1.5	960	24	600	620	540	540	600	600	570	590	680	680	680	680	560	560
14,000	8	1	NF	335	1.5	960	24	600	610	520	520	600	600	570	590	690	690	680	680	560	560
14,000	9	1	NF	325	2	960	24	560	560	520	520	550	550	520	540	630	630	660	670	520	520
14,000	10	2	SS	335	2	960	24	560	770	520	1600	560	1280	540	1300	630	630	660	830	520	1500
14,000	11	2	SS	335	2	960	24	560	820	520	1380	540	1560	520	1280	630	630	650	870	520	1500
14,000	12	1	NSSF	320	2	1025	28	620	650	580	580	620	620	520	540	700	700	730	730	560	560
14,000	13	1	SS	325	2	1030	28	620	750	580	1760	620	1110	570	1200	710	710	740	830	570	920
14,000	14	1	SS	330	2	1035	28	600	800	580	1540	610	1360	570	1120	700	700	740	850	570	1220
14,000	15	2	SS	335	2	1040	28	660	800	600	1460	630	1200	590	1080	730	730	750	820	580	1110
14,000	16	2	SS	330	1.5	1040	28	640	800	560	1580	640	1140	600	1110	740	740	750	820	580	1100
14,000	17	1	SS	336	1.5	1040	28	620	830	580	1580	640	1500	610	1130	740	740	760	870	580	1340
14,000	18	1	NF	335	1	1040	28	760	760	540	540	700	700	670	670	750	750	760	760	680	680
14,000	19	1	NF	335	1	1040	28	760	760	540	540	700	700	670	680	760	760	760	760	600	680
14,000	20	2	SS	335	1	1040	28	700	850	560	1600	700	1200	650	1120	750	750	760	850	620	1160

*NF - No Fire
NSSF - Non Self-Sustaining Fire
SS - Self-Sustaining Fire
+ - Temperature After Attempted
Fire Ignition

TABLE 3
AIR TEMPERATURE IN BEARING SUMP
SERIES 1 RUN 3

Shaft Speed (rpm)	Attempted Fire Ignition	Igniter Activated	Result	Oil Inlet Temp. (°F)	Oil Flow (gpm)	Hot Air Inlet Temp. (°F)	Air Flow Rate (scfm)	Mixture Temp. (°F) at Each Thermocouple Location													
								40°	40°+	70°	70°+	130°	130°+	180°	180°+	225°	225°+	315°	315°+	340°	340°+
7,000	1	2	*NF	300	2	955	28	700	700	640	640	660	660	650	650	720	720	710	710	660	660
10,000	2	2	NF	310	2	960	28	580	600	500	550	610	610	460	540	710	710	710	710	560	580
14,000	3	2	SS	315	2	955	28	580	800	580	1480	570	1540			620	620	700	870	520	1400
14,000	4	2	SS	325	2	880	28	570	760	520	1370	550	1370	540	1210	630	790	670	840	520	1240
14,000	5	2	NSSF	325	1.5	870	28	610	660	540	940	590	740	580	760	670	670	690	710	540	680
14,000	6	2	NSSF	330	1.5	870	28	610	680	540	1060	600	760	580	800	690	690	690	720	540	710
14,000	7	2	SS	325	1	870	28	690	710	560	1960	650	790	610	970	690	690	690	730	620	750
14,000	8	2	NF	325	1	865	28	690	690	550	550	650	650	620	620	695	695	690	690	630	630
14,000	9	2	NF	330	1	860	28	690	690	560	560	650	650	625	625	695	695	690	690	630	630

*NF - No Fire
 NSSF - Non-Self-Sustaining Fire
 SSF - Self-Sustaining Fire
 + - Temperature After Attempted
 Fire Ignition

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TABLE 4
AIR TEMPERATURE IN BEARING SUMP
SERIES 1 RUN 4

Shaft Speed (rpm)	Attempted Fire Ignition	Igniter Activated	Result	Oil Inlet Temp. (°F)	Oil Flow (gpm)	Hot Air Inlet Temp. (°F)	Air Flow Rate (scfm)	Mixture Temp.(°F) at Each Thermocouple Location											
								40°	40°+	70°	70°+	130°	130°+	180°	180°+	225°	225°+	340°	340°+
14,000	1	2	*NF	175	2	960	22	410	410	390	400	410	410	400	410	490	490	380	380
14,000	2	2	NF	205	2	980	23	440	460	420	420	450	450	430	440	530	530	410	410
14,000	3	2	NF	230	2	995	24	490	500	450	450	480	480	450	460	550	550	430	430
14,000	4	2	NF	260	2	1005	25	530	530	460	480	500	500	480	490	580	580	460	460
14,000	5	2	NSSF	295	2	1005	26	550	580	500	850	540	960	520	610	620	680	500	860
14,000	6	2	NSSF	295	2	1005	26	560	660	510	780	550	940	540	810			500	870
14,000	7	2	NSSF	290	1.5	1010	26	610	650	520	720	590	690	580	710	670	700	530	540
14,000	8	2	NSSF	290	1.5	1010	26	620	650	520	660	590	640	570	650			540	600

*NF - No Fire
NSSF - Non Self-Sustaining Fire
SSF - Self-Sustaining Fire
+ - Temperature After Attempted
Fire Ignition

TABLE 5
AIR TEMPERATURE IN OIL-FIRE PUMP
SERIES 1 RUN 5

Shaft Speed (rpm)	Attempted Fire Ignition	Igniter Activated	Result	Oil Inlet Temp. (°F)	Oil Flow (gpm)	Hot Air Inlet Temp. (°F)	Air Flow Rate (scfm)	Mixture Temp. (°F) at Each Thermocouple Location											
								40°	40°+	70°	70°+	130°	130°+	180°	180°+	225°	225°+	340°	340°+
14,000	1	2	*NSSF	330	2	990	5	510	540	480	520	480	480	470	530	540	570	460	480
14,000	2	2	NSSF	330	2	970	5	500	550	380	640	480	500	460	510	540	610	420	520
14,000	3	2	NSSF	360	1.5	885	4	520	560	480	740	500	520	480	530	550	600	500	520
14,000	4	2	NSSF	360	1.5	910	8	520	850	480	840	500	660	490	740			500	790
14,000	5	2	NSSF	370	1	820	8	540	650	480	740	520	590	500	610	590	740	520	580
14,000	6	2	NSSF	360	1	875	5	650	650	480	690	510	590			580	750	500	600
14,000	7	2	NSSF	320	1	900	7	520	770	460	800	460	650			550	820	480	670

*NF - No Fire
NSSF - Non Self-Sustaining Fire
SSF - Self-Sustaining Fire
+ - Temperature After Attempted
Fire Ignition

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FIGURE 9
MIXTURE TEMPERATURE AT THERMOCOUPLE LOCATIONS IN SUMP
Series 1 Run 1

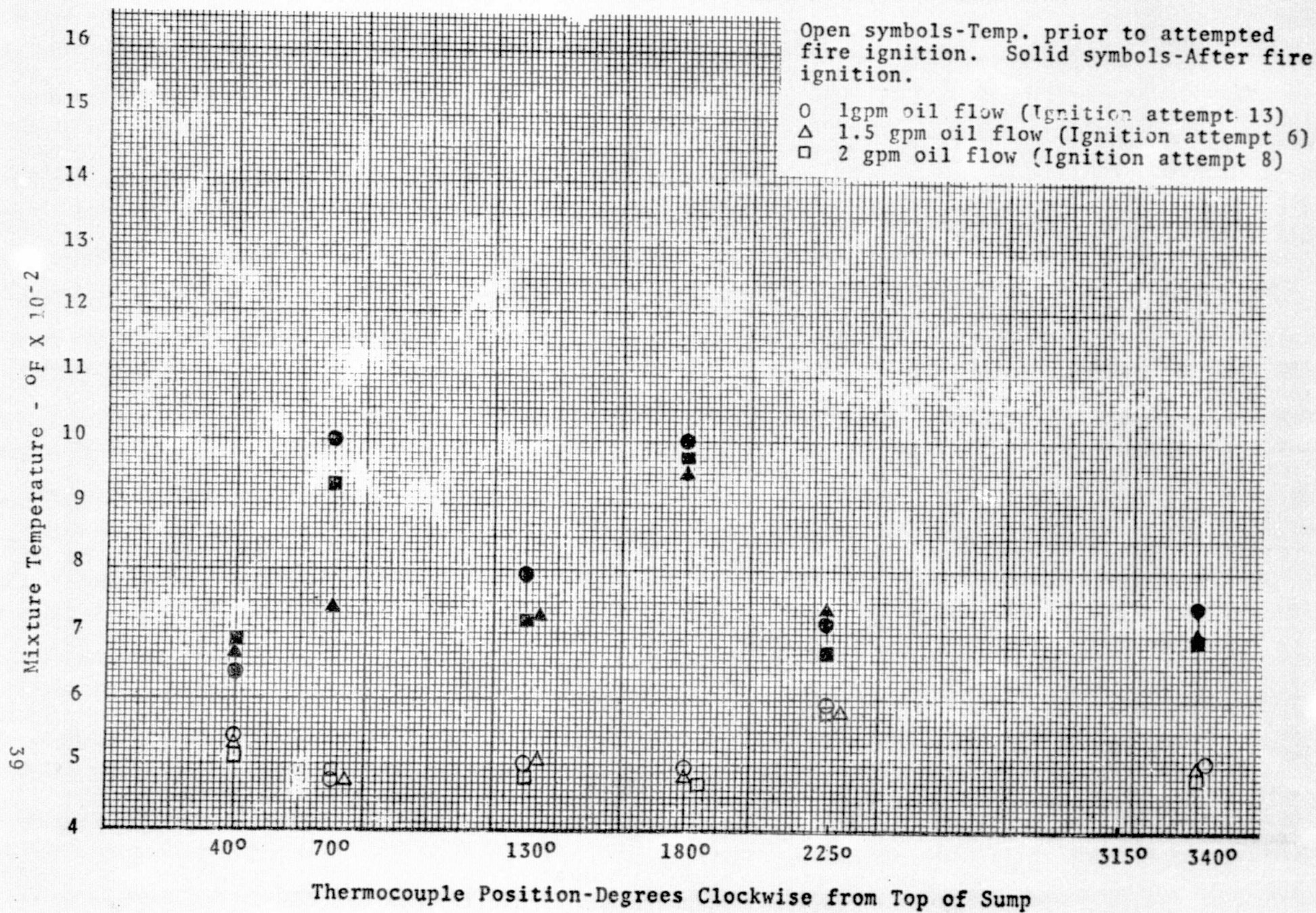


FIGURE 10
MIXTURE TEMPERATURE AT THERMOCOUPLE LOCATIONS IN SUMP
Series 1 Run 2

40

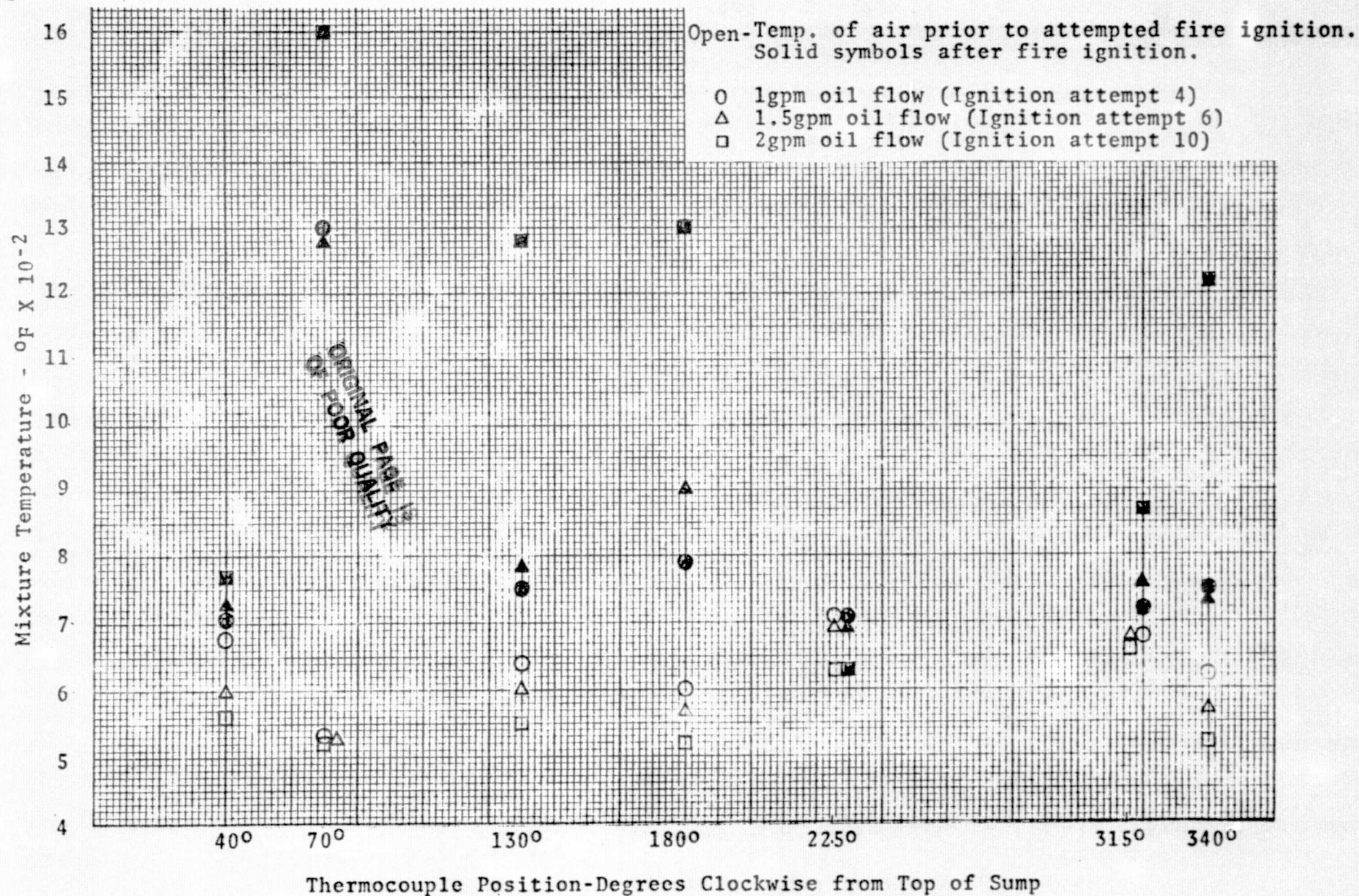


FIGURE 11
MIXTURE TEMPERATURE AT THERMOCOUPLE LOCATIONS IN SUMP
Series 1 Run 2 Continued

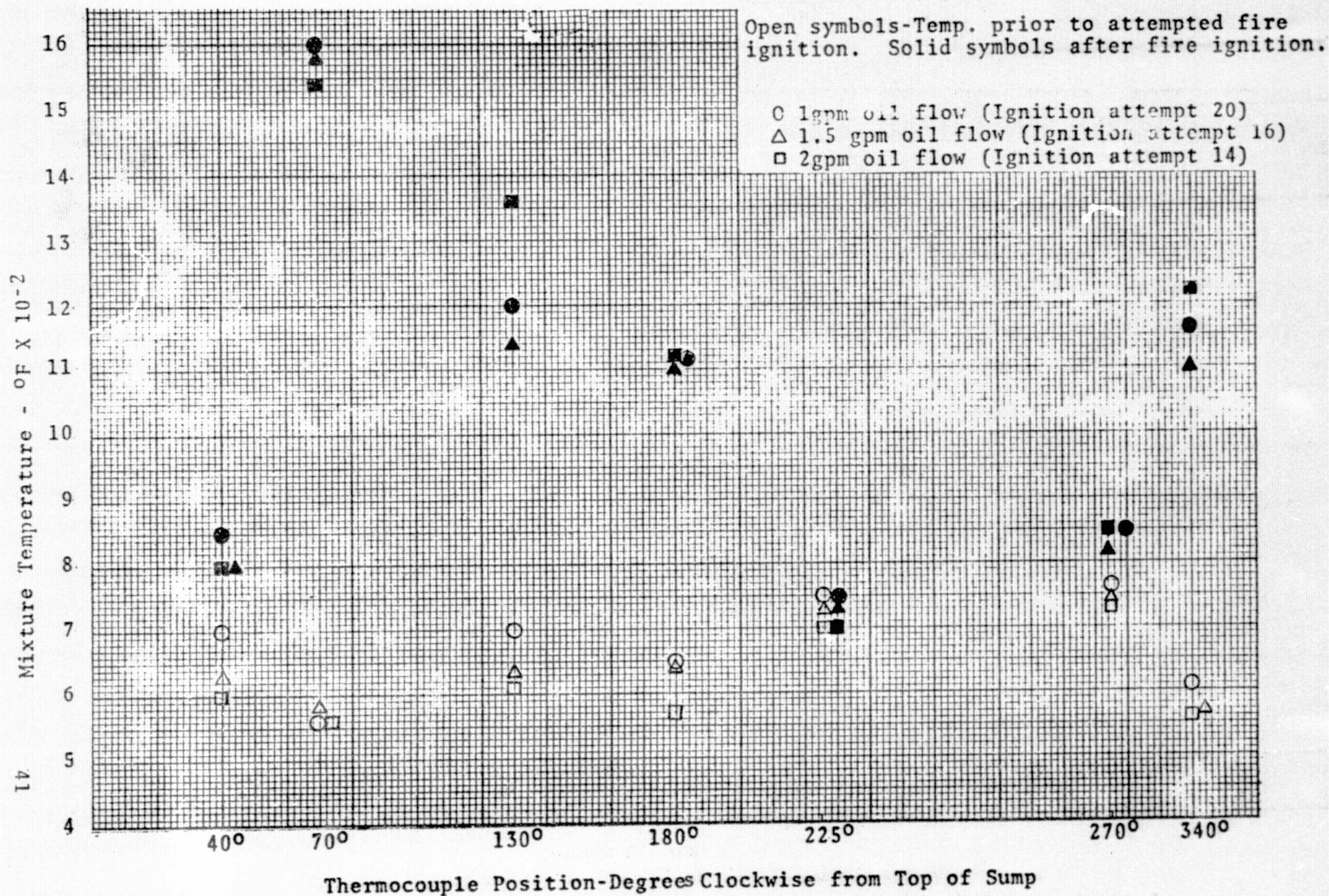
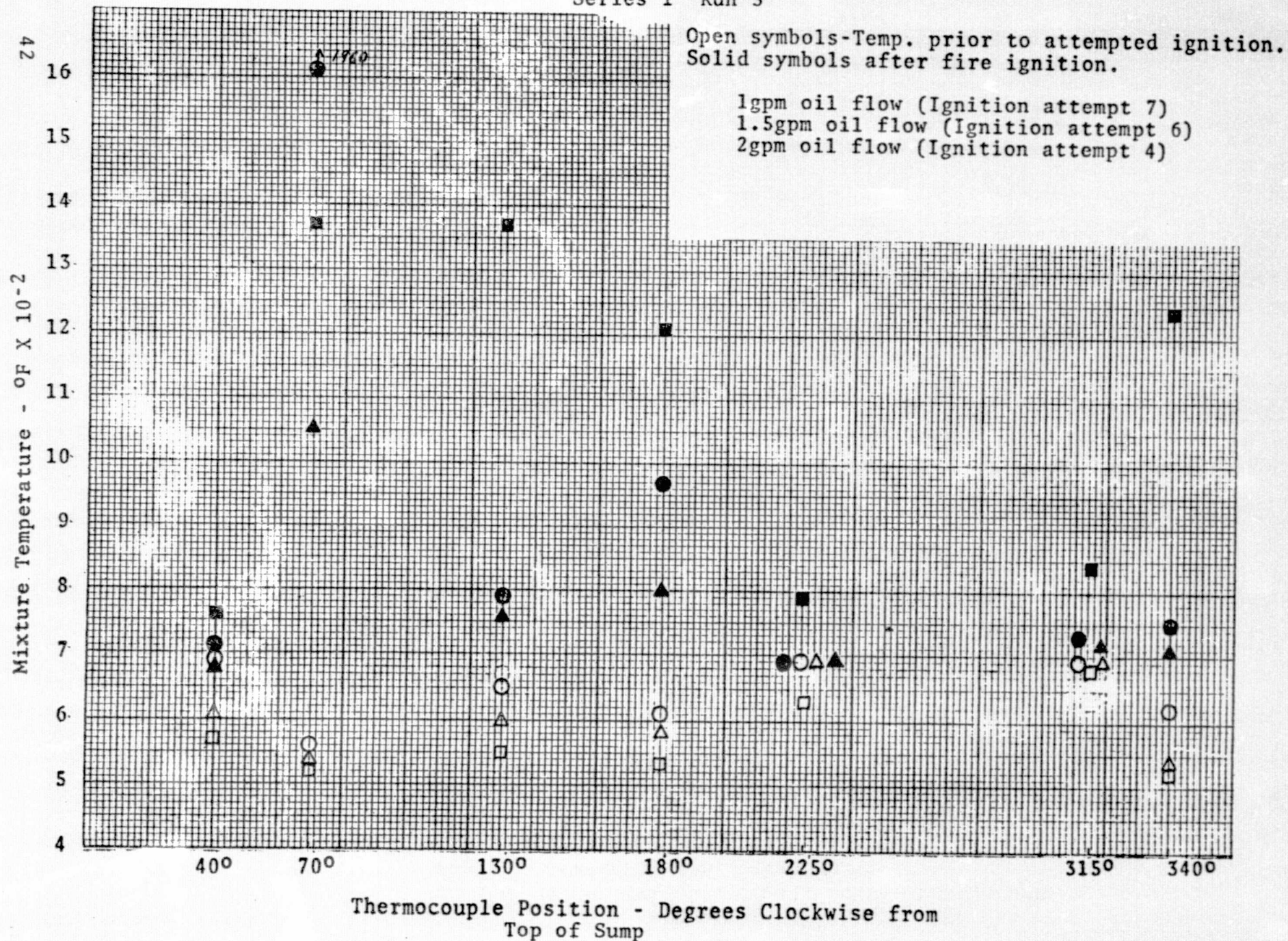


FIGURE 12
MIXTURE TEMPERATURE AT THERMOCOUPLE LOCATIONS IN SUMP
Series 1 Run 3



Reviewing the plotted data in conjunction with the tabulated data produced further understanding of what was taking place within the sump. In run 1, where the air inlet temperature was maintained between 725°K and 744°K (845°F and 880°F) with an air flow rate of $20.4 \text{ stdm}^3/\text{hr.}$ (12 scfm), the mixture temperature at ignitor 2 was only slightly above 525°K (485°F), the flash point of the oil. This would suggest that there was insufficient heat remaining in the air to evaporate additional oil and thus explain why only non self-sustaining fires could be ignited. It is also observed from Figure 9 that only slight changes occurred in the mixture temperature at all thermocouple points, including ignitor 2 location, due to changes in the oil flow rate. Thus it would appear that the mixture ratio did not change appreciably with oil flow rates in this run and that the ratio at ignitor 2 was sufficient to permit fire ignition with all oil flow rates used. At ignitor 1 it is assumed that the temperature was well above the flash point (no temperature data was obtained due to a faulty recorder). This assumption is based on the fact that in runs 2 and 3, the temperature at ignitor 1 was quite similar to that at the 225° azimuth location. This latter location was well above the fire point temperature of 558°K (545°F) throughout run 1. The low number of fires achieved with ignitor 1 leads one to consider that the mixture at this point was either too rich or too lean to burn. Since it must be concluded that there was little oil present for the heat in the air to transfer to and thus suggests that the mixture was too lean to support combustion.

In run 2 the air flow rate was increased to $40 \text{ stdm}^3/\text{hr.}$ (24 scfm) in the initial ignition attempts and $48 \text{ stdm}^3/\text{hr.}$ (28 scfm) in the latter attempts. It is observed from the plotted data that as the oil flow rate was increased the mixture temperature at all azimuth locations except 70° (air exhaust port) and 315° decreased. This suggests that more oil vapor was being generated, except at these two points, with the increased oil flow rates and explains why self-sustaining fires were ignited at ignitor 2 (130° azimuth) when the higher oil flow rates were supplied. At ignitor 1 (315° azimuth) the mixture temperature did not change appreciably with increased oil flow indicating that oil was not mixing with the air at this location and that the mixture was too lean to burn.

It is also observed from the plotted data that the self-sustaining fires spread more uniformly throughout the bearing sump than the non self-sustaining fires in run 1. However, the increases in temperature sensed at the position of ignitor 1 (315° azimuth), when self-sustaining fires were ignited at ignitor 2, were minimal which substantiated the previous conclusion that the mixture at this location was generally too lean to burn.

The plotted and tabulated data from run 3 substantiated the conclusions established from run 2 data; ie., increases in oil flow rate resulted in decreases in mixture temperatures at all points except 70° and 315°; and self-sustaining fires spread more uniformly throughout the sump. In addition, as previously discussed, the result of increased speed was the decrease in mixture temperature which suggested greater dispersion of the oil in the sump.

5.2 Test Series 2

Test series 2 was performed with a second configured baffle plate designed to minimize the mixing of the oil supplied to the bearing and the hot compressed air entering the bearing sump through a simulated seal leak. In essence, the baffle plate divided the bearing sump into two chambers connected only by a 0.36 mm (0.014 inch) gap between the baffle plate and the shaft sleeve. In addition, an oil flinger was attached to the shaft just inboard of the baffle plate to further reduce the passage of oil through the gap.

The purpose of this baffle was to determine if it were possible, without using another face seal or labyrinth seal in the bearing sump, to obtain a separation of the oil and air sufficient to generate a mixture too lean to burn. Although a too lean mixture would appear very difficult to obtain, due to the small amount of oil required in vapor form to produce a flammable mixture, the results of test series 1 previously discussed would suggest that possibility.

Series 2, Run 1

The first run was performed with conditions approximating those in Series 1, Run 1 with the major difference being the air inlet temperature which averaged approximately 22°K (40°F) higher in the second series. The extreme values of the test variables were:

Hot air flow rate through sump -	18.7-27.2 stdm ³ /hr. (11-16 scfm)
Hot air temperature entering hot air chamber -	744-772°K (880-930°F)
Oil inlet temperature -	432-466°K (320-380°F)
Oil flow rate -	0.23-0.46 m ³ /hr. (1-2 gpm)
Bearing outer ring temperature -	486-522°K (415-480°F)
Temperature at ignitor (mixture temperature) -	522-578°K (480-580°F)

A total of 22 attempts were made to ignite fires. In 19 of the 22 activations of the spark ignitor, both ignitors were used, non self-sustaining fires were ignited. However, seven of the fires were very mild producing only minor temperature increases even though the ignitor was activated for the full 60 seconds. The test data are presented in Table 6.

The major differences noted between this run and Series 1, Run 1 was that fires were readily ignited with both ignitors and the mixture temperature following fire ignition increased and decreased through several cycles during the 60 seconds of spark activation. This would suggest that the available oil vapor was being burned more rapidly than it was being generated. The test data showed that the mixture temperature at ignitor 1, when it was activated, was always above the flash point temperature of 525°K (485°F) except twice and in both cases fires were not ignited. An increase in the oil flow rate resulted in a decrease in the mixture temperature at the thermocouples as in Test 1, Run 1. This indicated, in addition to the fact that fires could be ignited, that oil was entering the hot air side of the baffle.

In general, essentially no improvement over the initial baffle configuration with respect to eliminating the ignition of fires by an electric spark was noted. Therefore, it was concluded that the new configuration did not sufficiently reduce the oil vapor air ratio, when a leakage air flow of 19-27 stdm³/hr. (11-16 scfm) was simulated, to a value which would prevent fires from being ignited. Either oil splashed through the opening between the baffle plate and shaft and/or the hot air flowed through the gap and returned with oil vapor of fine droplets which evaporated to produce vapor on the hot air side of the baffle.

Series 2, Run 2

The second test run was performed with conditions similar to those applied in Series 1, Run 2. The extreme values of the test variables were:

TABLE 6
TEST DATA
SERIES 2 RUN 1

Ignition Attempt (No.)	Results	Shaft Speed (rpm)	Spark Ignitor No.	Air Flow Rate (scfm)	Air In Temp. (°F)	Oil Flow Rate (gpm)	Oil In Temp. (°F)	Brg. Temp. (°F)	Mixture Temp. At Ignitor (°F)
1	* NSSF	14,000	1	13	880	1.5	340	450	520
2	NSSF	14,000	1	13	885	1.5	350	450	520
3	NSSF	14,000	1	13	890	1.5	360	455	520
4	NSSF	14,000	2	13	905	1.5	370	455	550
5	NSSF	14,000	2	13	910	1.5	380	430	520
6	NSSF	14,000	2	13	910	1.5	365	430	540
7	NSSF	14,000	2	16	910	2	320	420	520
8	NSSF	14,000	2	16	910	2	330	430	540
9	NSSF	14,000	1	16	930	2	340	415	500
10	NSSF	14,000	1	13	930	2	350	415	500
11	NSSF	14,000	1	13	930	1.5	330	430	520
12	NSSF	14,000	1	13	930	1.5	325	430	540
13	NSSF	14,000	1	13	930	1	330	480	560
14	NSSF	14,000	1	13	930	1	330	470	580
15	NSSF	14,000	2	13	920	1	340	470	520
16	NSSF	14,000	2	15	925	1	350	470	580
17	NF	14,000	2	11	935	2	335	450	520
18	NF	14,000	2	11	930	2	340	440	500
19	NF	14,000	1	11	925	2	360	430	480
20	NSSF	14,000	1	11	880	1.5	360	440	480
21	NSSF	14,000	2	11	880	1.5	330	450	500
22	NSSF	14,000	1	11	870	1.5	340	470	520

*NF - No Fire

NSSF - Non-Self-Sustaining Fire

SSF - Self-Sustaining Fire

Hot air flow rate through sump -	40.8-49.3 stdm ³ /hr. (24-29 scfm)
Hot air temperature entering hot air chamber -	783-868 ^o K (950-1050 ^o F)
Oil inlet temperature -	450-464 ^o K (345-375 ^o F)
Oil flow rate -	0.23-0.57 m ³ /hr. (1-2.5 gpm)
Bearing outer ring temperature -	494-538 ^o K (430-510 ^o F)
Temperature at ignitor (mixture temperature) -	632-656 ^o K (680-720 ^o F)

Series 2 Summary

In only seven of the 27 activations of the spark ignitors were fires ignited (both ignitors were used and fires were started by both). All fires ignited were non self-sustaining and in the two ignitions where the ignitor was left on after the fires started, the temperatures fluctuated up and down indicating the fires were going out and then reigniting as in Series 2, Run 1. The fewer and less severe fires was a considerable improvement over that obtained in Series 1, Run 2 where eight self-sustaining and four non self-sustaining fires were ignited in 20 activations. Another major difference between the two tests was the appreciably higher mixture temperatures measured at the ignitor locations in this test, see Table 7. These higher temperatures indicated that little oil was mixing with the air passing the ignitors; thus, reducing the heat transfer from the air. This conclusion is further substantiated by the fact that increases in the oil flow rate from 0.34 to 0.57 m³/hr. (1.5 to 2.5 gpm) had essentially no effect on the mixture temperatures measured at the ignitors. However, this was not true at all thermocouple locations within the sump. The temperatures measured at the thermocouples located at 40^o and 70^o did vary as expected with oil flow rate changes (increased with decreased oil flow and decreased with increased oil flow). Thus it was concluded from this test that although it appeared that the change in baffle configuration was beneficial in producing conditions with the high air flow that were less conducive to fire being ignited it was only a zone or area influence. At other locations, other than where the ignitors were positioned, it is likely that fires could have more readily been ignited.

TABLE 7
TEST DATA
SERIES 2 RUN 2

<u>Ignition Attempt (No.)</u>	<u>Results</u>	<u>Shaft Speed (rpm)</u>	<u>Spark Ignitor No.</u>	<u>Air Flow Rate (scfm)</u>	<u>Air In Temp. (°F)</u>	<u>Oil Flow Rate (gpm)</u>	<u>Oil In Temp. (°F)</u>	<u>Brg. Temp. (°F)</u>	<u>Mixture Temp. At Ignitor (°F)</u>
1	*NSSF	14,000	2	29	970	2	345	430	710
2	NSSF	14,000	2	29	960	2	345	450	720
3	NSSF	14,000	1	29	960	2	345	440	700
4	NSSF	14,000	1	29	960	2	345	440	680
5	NF	14,000	1	29	950	1.5	345	490	700
6	NF	14,000	1	29	950	1.5	350	490	700
7	NF	14,000	2	29	950	1.5	365	500	700
8	NF	14,000	2	29	950	1.5	370	500	700
9	NF	14,000	2	29	950	2	375	480	700
10	NF	14,000	1	29	950	2	350	460	700
11	NF	14,000	1	29	1000	2	340	450	700
12	NF	14,000	2	29	1010	2	350	455	700
13	NF	14,000	2	24	1010	2	370	465	700
14	NSSF	14,000	1	24	1010	2	340	460	700
15	NSSF	14,000	1	24	1010	2	340	455	710
16	NSSF	14,000	1	24	1010	2	350	460	710
17	NF	14,000	1	24	1010	1.5	360	490	720
18	NF	14,000	1	24	1010	1.5	365	500	720
19	NF	14,000	2	24	1010	1.5	365	510	700
20	NF	14,000	2	24	1010	2	340	470	700
21	NF	14,000	2	24	1010	2	340	460	700
22	NF	14,000	1	24	1010	2	340	430	720
23	NF	14,000	2	24	1010	2.5	350	430	720
24	NF	14,000	1	24	1030	2	360	480	720
25	NF	14,000	2	24	1030	2	350	470	720
26	NF	14,000	2	28	970	2	370	480	720
27	NF	14,000	2	28	950	1	370	480	720

*NF - No Fire

NSSF - Non-Self-Sustaining Fire

SSF - Self-Sustaining Fire

These results suggest the possibility of designing a sump which would control the flow of air and oil (direct the flow paths) in a manner which would maintain a mixture too lean at a possible ignition source to support combustion. The use of thermocouples strategically positioned in the sump to measure mixture temperature changes with oil flow, air flow and shaft speed changes could be considered as a practical means of determining the effectiveness of the design.

The two tests performed with the second baffle design showed that the configuration was not effective in eliminating conditions that were conducive to flammability, but suggested that proper baffling could possibly be effective.

Evaluating the input parameters for the first two test series as criteria for determining flammability conditions within the sump, as was the goal of the program, several conclusions could be established. Fires could be ignited over the full range of air flow rates evaluated 7 to 49 stdm³/hr. (4 to 29 scfm) and over the complete range of inlet air temperatures 724° to 833°K (845 to 1040°F) evaluated. However, the greater the air flow rate the more severe the fires. At the low air flow rates of 7 to 14 stdm³/hr. (4 to 8 scfm) only minor non self-sustaining fires were ignited. With air flow rates of 20 to 27 stdm³/hr. (12 to 16 scfm) the fires were also non self-sustaining, but much higher mixture temperatures increases occurred. With air flow rates of 41 to 49 stdm³/hr. (24 to 29 scfm) self-sustaining fires were ignited with the first baffle configuration, which spread more generally throughout the sump with temperature increases as high as 811°K (1000°F) before the fires were extinguished. This would suggest that the greater flow rates resulted in greater dispersion of the oil and greater heat transfer from the air to the oil; thus, generating more oil vapor. It should be noted that, even though fires were ignited over the full air flow and temperature ranges, fires could not always be started and the changes that resulted in this condition could not always be determined.

Fires were also ignited over the full range of input oil flow rates evaluated 0.23 to 0.46 m³/hr. (1 to 2 gpm). Again it is noted that with the oil flow rates in this range, fires were not started in all attempts. In several cases, fires could not be ignited with an oil input of 0.23 m³/hr. (1 gpm) but were ignited when the flow was increased to 0.34 and 0.46 m³/hr. (1.5 and 2 gpm). This suggested that an increase in oil flow also produced a greater dispersion of the oil which resulted

in more oil vapor being generated. This assumption was substantiated by the decrease in the mixture temperature at the ignitor, greater heat transfer from the air to the oil, in most cases where an increase in oil flow rates resulted in a fire ignition.

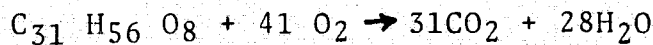
Oil inlet temperature was the only input parameter which produced results from which more than a general trend could be established even in a specific sump configuration. With the inlet air flow rate and temperature at 44 stdm³/hr. and 814°K (26 scfm and 1005°F) which approached the most severe conditions tested, no fires could be ignited until the oil inlet temperature was increased to 420°K (295°F). This observation indicates that conditions susceptible to fire ignitions can be prevented by maintaining a low oil inlet temperature even when high hot air leakage rates into the sump are present.

To further evaluate the test data, a check was performed to determine the correlation between the ignition of fires and the mixture temperature at the ignitor relative to the flammability limits. The flammability limits would provide a much less complex and easily measured judgement criterion (compared to the combination of the inlet parameters which may be very dependent on sump configuration) if good correlation existed.

As stated in reference 4, the ability of a liquid fuel to form flammable vapor-air mixture is defined by its temperature and concentration limits of flammability. The lower temperature limit (LL) is realized when the liquid fuel temperature is high enough to produce a minimum fuel vapor concentration which when uniformly mixed with air will sustain flame, if ignited by an external heat source. This temperature limit is usually slightly lower than the flash point of the liquid. The upper temperature limit (UL) corresponds to the liquid fuel temperature above which the equilibrium fuel concentration of the saturated vapor-air mixture is too rich to sustain flame.

The concentration of fuel vapor is determined by its equilibrium vapor pressure at any given temperature. The equilibrium fuel-air ratio is therefore the ratio of the vapor pressure of the fuel to the air pressure in the chamber.

To establish the flammability limits for the lubricant used in the program (Type II Ester, meeting MIL-L-23699 specifications), the stoichiometric ratio (C_S) was determined and then the limits calculated using equations presented in Section 1.2. To establish the stoichiometric ratio the chemical equations $C_{31}H_{56}O_8$ was chosen to represent a typical molecule of the lubricant and has a molecular weight of 556 grams/mol. Using this equation and assuming complete oxidation where the products are carbon dioxide and water, the following balanced equation can be written.

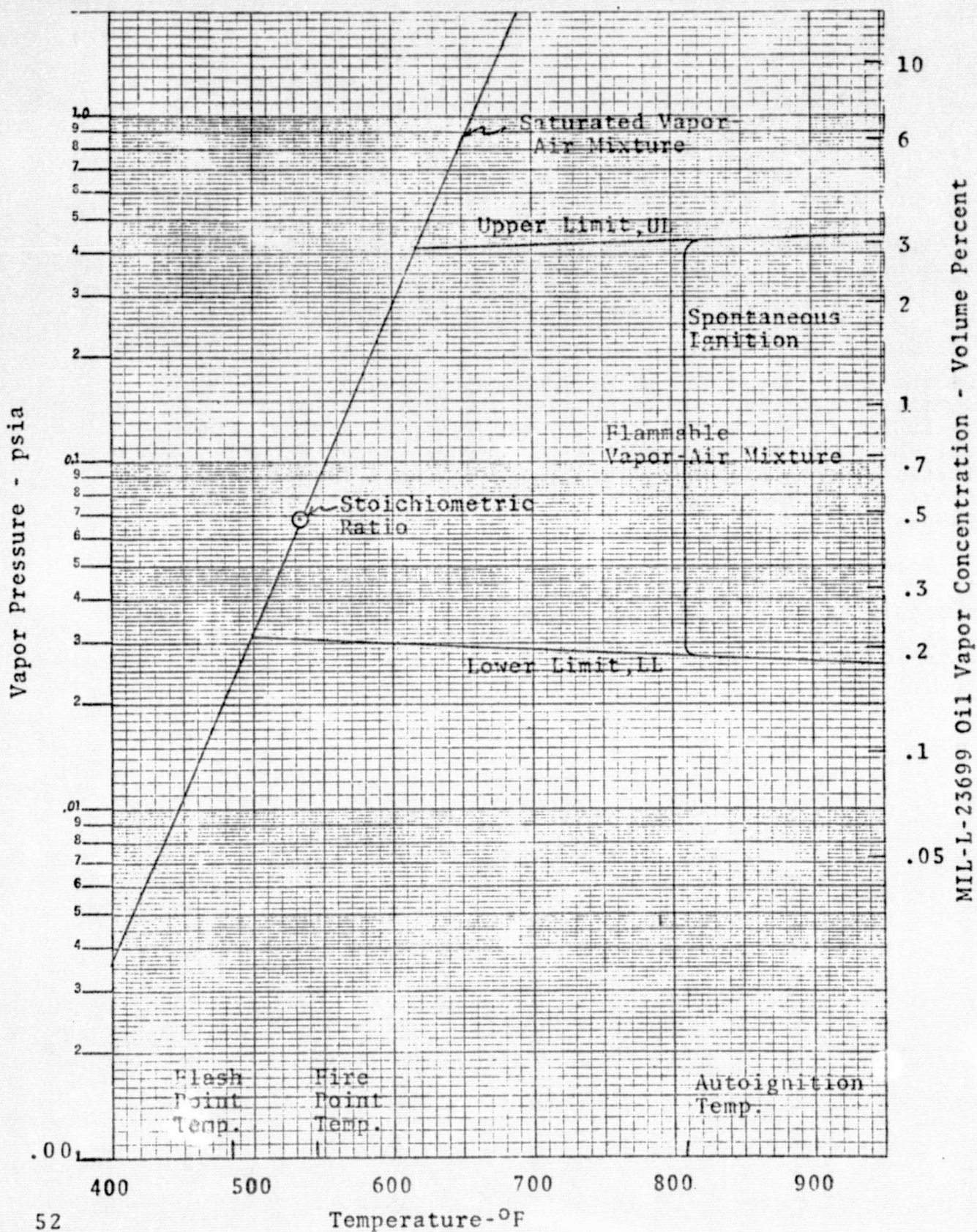


Thus the molar ratio at the stoichiometric point is 1/41 and the molar ratio when combining with air (1 mol of O_2 in 5 mol of air) is $1/5 \times 1/41 = 0.0049 = 0.49\%$ oil vapor to air by volume. Using the vapor pressure versus temperature curve for MIL-L-23699 lubricants from reference 7 and the calculated flammability limits, the flammability diagram presented in Figure 13 was established.

From the flammability diagram a UL of $600^\circ K$ ($620^\circ F$) and a LL of $533^\circ K$ ($500^\circ F$) were selected for comparison values. Comparison of these limits with the mixture temperatures measured at the actuated ignitors in the first two test series resulted in the following statistical data:

Test Series (No.)	Run (No.)	Mixture Temp. Below LL		Mixture Temp. Between Limits		Mixture Temp. Above UL	
		<u>Ignition Attempts</u>	<u>No. of Fires</u>	<u>Ignition Attempts</u>	<u>No. of Fires</u>	<u>Ignition Attempts</u>	<u>No. of Fires</u>
1	1	1	0	8	8	0	0
1	2	0	0	5	5	15	6
1	3	0	0	4	4	5	1
1	4	3	0	5	4	0	0
1	5	1	1	6	6	0	0
2	1	1	0	21	19	0	0
2	2	0	0	0	0	27	7
TOTAL		6	1	49	46	47	14

FIGURE 13
FLAMMABILITY DIAGRAM OF MIL-L-23699 FLUID



These data show a reasonably good correlation with the flammability limits. This is especially true at mixture temperatures between the limits where fires would be expected (46 fires ignited in 49 attempts) and also below the lower limit where fires would not be expected (1 fire ignited in 6 attempts). At mixture temperatures above the upper limit, where fires would also not be expected, the poorest correlation existed (14 fires ignited in 47 attempts).

The result of this evaluation would certainly suggest the feasibility of using the mixture temperature measurement in a sump as a reasonable method of judging the flammability conditions. However, the relatively low percentage of fires ignited above the UL would have to be considered to be the result of a too rich mixture (excessive amount of oil vapor). This is contrary to the conclusions drawn by attempting to evaluate what was occurring in the sump. In that case, as previously discussed, when high mixture temperatures were measured, it was considered to be the result of a lean mixture (insufficient oil vapor produced to provide a flammable mixture), i.e. the temperature was high because there was little heat transferred to oil, thus an equilibrium state did not develop. A review of the tabulated data also showed some of the most severe fires (self-sustaining) occurred when the mixture temperature was 56° to 112°K (100° to 200°F) above the expected limiting temperature of 600°K (620°F). This could be expected, however, if an equilibrium temperature condition did not exist, (oil temperature had not reached the air or measured temperature), which would decrease the amount of oil vapor formed and thus result in a leaner mixture. This would also suggest, however, that there should have been several cases, when the temperatures were between the limits, that fires would not ignite since the mixture would have been too lean.

Because of these apparent discrepancies, and the fact that only a few cases had been evaluated with mixture temperatures below the LL, Test Series 3 and 4 were performed to obtain additional data to substantiate the possibility of using the mixture temperature as a flammability judgement criterion.

5.3 Test Series 3

Since only six attempts had been made to ignite fires during the first two series when the measured mixture temperature at the ignitors was below the LL, all three runs in Test Series 3 were initiated with the mixture temperature below the LL. This condition was maintained by inserting a low oil and/or low air temperature. The mixture temperature was then increased to values between the limits by increasing either or both the oil and air temperatures.

Series 3, Run 1

The first test run was performed with the relatively low oil temperature of approximately 360°K (190°F) and a high air flow rate value of 43 to 48 $\text{stdm}^3/\text{hr.}$ (25 to 28 scfm). The extreme value of the test variables were:

Hot air flow rate through sump	- 43 to 48 $\text{stdm}^3/\text{hr.}$ (25-28 scfm)
Hot air temperature entering hot air chamber	- 655-775 $^{\circ}\text{K}$ (720-935 $^{\circ}\text{F}$)
Oil inlet temperature	- 352-368 $^{\circ}\text{K}$ (175-200 $^{\circ}\text{F}$)
Oil flow rate	- 0.23-0.57 $\text{m}^3/\text{hr.}$ (1-25 gpm)
Bearing outer ring temperature	- 409-437 $^{\circ}\text{K}$ (275-325 $^{\circ}\text{F}$)
Temperature at ignitor (mixture temperature)	- 516-593 $^{\circ}\text{K}$ (470-610 $^{\circ}\text{F}$)

The test run was initiated with an oil flow rate of 0.34 $\text{m}^3/\text{hr.}$ (1.5 gpm) and temperature of 358 $^{\circ}\text{F}$ and an air flow rate and temperature of 48 $\text{stdm}^3/\text{hr.}$ and 635 $^{\circ}\text{K}$ (28 scfm and 720 $^{\circ}\text{F}$) respectively. The oil flow rate was changed to 0.23, 0.46, and 0.57 $\text{m}^3/\text{hr.}$ (1, 2, and 2.5 gpm) with each spark ignitor activated at least once for 60 seconds at each flow rate. With these conditions imposed, the mixture temperatures at the ignitors varied from 517 to 543 $^{\circ}\text{K}$ (470 to 520 $^{\circ}\text{F}$) which is from 17 $^{\circ}\text{K}$ (30 $^{\circ}\text{F}$) below to 11 $^{\circ}\text{K}$ (20 $^{\circ}\text{F}$) above the LL. In none of the 14 ignition attempts were fires ignited, see Table 8.

To evaluate the possibility of fire igniting at higher mixture temperatures, the air inlet temperature was increased in two incremental steps, first to 719 $^{\circ}\text{K}$ (835 $^{\circ}\text{F}$) and then to 771 $^{\circ}\text{K}$ (930 $^{\circ}\text{F}$), while maintaining the same air flow rate and oil inlet temperature. At each new temperature condition, the oil flow rates were varied in 0.12 $\text{m}^3/\text{hr.}$ (.5 gpm) steps from 0.23 to 0.57 $\text{m}^3/\text{hr.}$ (1 to 2.5 gpm) and ignition attempts made at each flow rate. No fire ignitions occurred under any of the conditions imposed even though the mixture temperature increased to a maximum value of 593 $^{\circ}\text{K}$ (610 $^{\circ}\text{F}$) which is well above the LL of 533 $^{\circ}\text{K}$ (500 $^{\circ}\text{F}$), but still below the UL.

These results, contrary to prior test results where fires were ignited in 94 percent of the cases when the mixture temperature was between the LL and UL, indicated that the flammability temperature limits may not be as reliable of a judgement criterion as initially considered. However, it was noted that the oil inlet temperature was maintained within 3 $^{\circ}\text{K}$ (5 $^{\circ}\text{F}$) of the minimum value where fires had previously been ignited.

TABLE 8
TEST DATA
SERIES 3 RUN 1

Ignition Attempt (No.)	Results	Shaft Speed (rpm)	Spark Ignitor No.	Air Flow Rate (scfm)	Air In Temp. (°F)	Oil Flow Rate (gpm)	Oil In Temp. (°F)	Brg. Temp. (°F)	Mixture Temp. At Ignitor (°F)
1	*NF	14,000	1	28	720	1.5	185	295	480
2	NF	14,000	2	28	720	1.5	185	295	480
3	NF	14,000	2	26	720	1.5	190	310	480
4	NF	14,000	1	28	720	1.5	190	310	480
5	NF	14,000	1	28	730	1	190	360	510
6	NF	14,000	2	28	740	1	180	345	510
7	NF	14,000	2	28	740	1	180	345	520
8	NF	14,000	1	27	740	1	180	355	520
9	NF	14,000	1	27	740	2	180	300	500
10	NF	14,000	2	27	740	2	180	295	500
11	NF	14,000	2	27	745	2.5	185	285	495
12	NF	14,000	1	27	745	2.5	185	285	485
13	NF	14,000	1	27	835	2.5	175	280	510
14	NF	14,000	2	26	830	2.5	180	275	520
15	NF	14,000	2	26	835	2	185	300	540
16	NF	14,000	1	26	835	2	185	300	540
17	NF	14,000	1	27	840	1.5	185	305	560
18	NF	14,000	2	27	840	1.5	185	305	550
19	NF	14,000	2	25	840	1	175	335	560
20	NF	14,000	2	25	840	1	175	350	560
21	NF	14,000	1	25	840	1	175	380	580
22	NF	14,000	1	27	900	2.5	200	295	520
23	NF	14,000	2	26	900	2.5	200	295	540
24	NF	14,000	2	26	915	2	190	300	560
25	NF	14,000	1	25	920	2	190	300	540
26	NF	14,000	1	25	930	1.5	185	310	580
27	NF	14,000	2	25	930	1.5	185	310	585
28	NF	14,000	1	25	935	1.25	175	325	600
29	NF	14,000	2	25	935	1.25	175	325	610

*NF - No Fire
NSSF - Non-Self-Sustaining Fire
SSF - Self-Sustaining Fire

Series 3, Run 2

Run 2 was also performed with an air flow rate of 48 stdm³/hr. (28 scfm) and initiated with oil inlet and air inlet temperatures which resulted in the mixture temperature being below the LL, see Table 9. The oil inlet temperature was then increased in incremental steps from the initial value of 368°K (200°F) to 432°K (320°F) while maintaining the air inlet temperature essentially constant at 655°K (720°F) and varying the oil flow rates from 0.23 to 0.57 m³/hr. (1 to 2.5 gpm). The air inlet temperature was then increased in two steps; first to approximately 728°K (850°F) and then to 811°K (1000°F) while maintaining the oil inlet temperature between 421 and 438°K (300 and 330°F). The oil temperature was then increased to approximately 452°K (355°F) and the air temperature to 819°K (1015°F). The extreme values of the test variables were:

Hot air flow rate through sump	- 48 stdm ³ /hr. (28 scfm)
Hot air temperature entering hot air chamber	- 652-819°K (715-1015°F)
Oil inlet temperature	- 387-455°K (200-360°F)
Oil flow rate	- 0.23-0.57 m ³ /hr. (1-2.5 gpm)
Bearing outer ring temperature	- 414-489°K (285-420°F)
Temperature at ignitor (mixture temperature)	- 510-636°K (460-685°F)

In none of the first 25 activation of spark ignitors was a fire ignited even though the mixture temperature increased from 520 to 583°K (480 to 590°F) and the oil inlet temperature was as high as 455°K (360°F). Not until the mixture temperature had reached 588°K (600°F) at ignitor 1 and 635°K (685°F) at ignitor 2 were fires ignited. All six fires ignited were very minor as evidenced by a small increase in mixture temperature and the fires would go out even with the ignitor still activated. It is also noted that fires could not be ignited until the oil temperature was above 432°K (320°F) which was the minimum value in all but one prior test. Thus, the results again showed poor correlation with the flammability limits.

Series 3, Run 3

In Test Series 2 where the more restrictive baffle was initially used (also used in Test Series 3 and 4) fewer fires were ignited with the higher air flow; thus, the third run was performed with the lower air flow rate of 27 stdm³/hr. (16 scfm). The test was initiated with an oil inlet temperature of 380°K (225°F) and air temperature of 649°K (710°F), see Table 10. The extreme values of the test variable were:

TABLE 9
TEST DATA
SERIES 3 RUN 2

Ignition Attempt (No.)	Results	Shaft Speed (rpm)	Spark Ignitor No.	Air Flow Rate (scfm)	Air In Temp. (°F)	Oil Flow Rate (gpm)	Oil In Temp. (°F)	Brg. Temp. (°F)	Mixture Temp. At Ignitor (°F)
1	*NF	14,000	1	28	715	2	200	285	460
2	NF	14,000	2	28	715	2	200	285	470
3	NF	14,000	2	28	715	1.5	215	300	465
4	NF	14,000	1	28	715	1.5	215	300	490
5	NF	14,000	1	28	715	1.5	250	350	490
6	NF	14,000	2	28	715	1.5	250	350	520
7	NF	14,000	2	28	715	2	250	340	520
8	NF	14,000	1	28	715	2	250	340	520
9	NF	14,000	1	28	720	1.5	290	370	510
10	NF	14,000	2	28	720	1.5	290	370	510
11	NF	14,000	2	28	720	2	310	365	530
12	NF	14,000	1	28	720	2	310	365	530
13	NF	14,000	1	28	715	1	310	385	525
14	NF	14,000	2	28	715	1	310	385	525
15	NF	14,000	1	28	720	1	325	395	545
16	NF	14,000	1	28	720	2	330	385	520
17	NF	14,000	2	28	720	2	330	385	520
18	NF	14,000	2	28	720	2.5	315	360	520
19	NF	14,000	1	28	720	2.5	320	360	520
20	NF	14,000	1	28	840	2	305	375	540
21	NF	14,000	2	28	840	2	305	375	540
22	NF	14,000	2	28	860	1	315	390	585
23	NF	14,000	1	28	860	1	315	400	585
24	NF	14,000	1	28	865	1.5	325	395	580
25	NF	14,000	2	28	865	1.5	330	395	590
26	NF	14,000	2	28	1000	1.5	300	380	645
27	NF	14,000	1	28	1000	1.5	300	380	600
28	NF	14,000	1	28	1010	1	305	405	630
29	NF	14,000	1	28	1010	1	330	425	650
30	NF	14,000	2	28	1015	1	332	420	685
31	NF	14,000	2	28	1015	2	350	415	685
32	NF	14,000	1	28	1015	2	360	420	630

*NF - No Fire

NSSF - Non-Self-Sustaining Fire

SSF - Self-Sustaining Fire

TABLE 10
TEST DATA
SERIES 3 RUN 3

Ignition Attempt (No.)	Results	Shaft Speed (rpm)	Spark Ignitor No.	Air Flow Rate (scfm)	Air In Temp. (°F)	Oil Flow Rate (gpm)	Oil In Temp. (°F)	Brg. Temp. (°F)	Mixture Temp. At Ignitor (°F)
1	*NF	14,000	1	16	710	2	225	300	440
2	NF	14,000	2	16	710	2	225	300	450
3	NF	14,000	2	16	720	1	225	335	460
4	NF	14,000	1	16	720	1	225	335	460
5	NF	14,000	1	16	725	1.5	240	320	460
6	NF	14,000	2	16	725	1.5	240	320	460
7	NF	14,000	2	16	840	1.5	230	315	485
8	NF	14,000	1	16	865	1.5	230	315	480
9	NF	14,000	1	16	870	1	235	330	500
10	NF	14,000	2	16	870	1	235	330	505
11	NF	14,000	2	16	875	2	225	310	505
12	NF	14,000	1	16	875	2	225	310	500
13	NF	14,000	1	16	890	2	310	370	540
14	NF	14,000	2	16	890	2	310	370	540
15	NF	14,000	2	16	895	1	315	395	550
16	NF	14,000	1	16	895	1	315	395	550
17	NF	14,000	1	16	895	1.5	325	390	550
18	NF	14,000	2	16	895	1.5	325	390	560

*NF - No Fire
NSSF - Non-Self-Sustaining Fire
SSF - Self-Sustaining Fire

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Hot air flow rate through sump	- 27 stdm ³ /hr. (16 scfm)
Hot air temperature entering	
hot air chamber	- 649-753°K (710-895°F)
Oil inlet temperature	- 380-337°K (225-325°F)
Oil flow rate	- 0.23-0.46 m ³ /hr. (1-2 gpm)
Bearing outer ring temperature	- 400-455°K (260-360°F)
Temperature at ignitor	
(mixture temperature)	- 500-566°K (440-560°F)

With no fire ignited in the first six attempts where the mixture temperature was below the LL, the air temperature was increased to 738°K (870°F) to increase the mixture temperature to within the limits. Still no fires could be ignited and the oil temperature was increased to approximately 436°K (325°F). Again with the mixture temperature well within the flammability limits, no fires were ignited.

5.4 Test Series 4

Test Series 4, which consisted of three test runs, was performed to evaluate the possibility of igniting fires when the oil inlet temperature values were maintained in a high range and the hot air inlet values were relatively low; thus, obtaining a mixture temperature either slightly below the LL or between the limits. The extreme values of the test variables were:

Hot air flow rate through sump	- 27 stdm ³ /hr. (16 scfm)
Hot air temperature entering	
hot air chamber	- 652-686°K (715-775°F)
Oil inlet temperature	- 430-458°K (315-365°F)
Oil flow rate	- 0.23-0.57 m ³ /hr. (1-2.5 gpm)
Bearing outer ring temperature	- 460-488°K (370-420°F)
Temperature at ignitor	
(mixture temperature)	- 522-539°K (480-510°F)

Series 4, Run 1

The first run was initiated with an oil inlet temperature of 430°K (315°F) and flow rate of 0.46 m³/hr. (2 gpm), and air temperature of 652°K (715°F). An air flow rate of 27 stdm³/hr. (16 scfm) was maintained throughout the run. The initial inlet conditions resulted in a mixture temperature of 522°K (480°F) at ignitor 1 during the first two activations. Non-self-sustaining fires were ignited in both cases. No fires were ignited with ignitor 2 in essentially similar conditions with oil flow rates of 0.23 and 0.46 m³/hr. (1 and 2 gpm), see Table 11.

TABLE 11
TEST DATA
SERIES 4 RUN 1

Ignition Attempt (No.)	Results	Shaft Speed (rpm)	Spark Ignitor No.	Air Flow Rate (scfm)	Air In Temp. (°F)	Oil Flow Rate (gpm)	Oil In Temp. (°F)	Brg. Temp. (°F)	Mixture Temp. At Ignitor (°F)
1	*NSSF	14,000	1	16	715	2	315	370	480
2	NSSF	14,000	1	16	715	2	315	370	480
3	NF	14,000	2	16	710	2	330	380	490
4	NF	14,000	2	16	710	1	340	400	510
5	NF	14,000	1	16	710	1	340	400	550
6	NSSF	14,000	1	16	710	1.5	345	405	530
7	NF	14,000	2	16	710	1.5	345	405	510
8	NF	14,000	2	16	710	2	340	390	500
9	NSSF	14,000	1	16	710	2	340	390	500
10	NSSF	14,000	1	16	710	2.5	360	390	480
11	NSSF	14,000	2	16	710	2.5	360	390	480
12	NSSF	14,000	2	16	750	1.5	360	420	520
13	NSSF	14,000	1	16	750	1.5	360	420	550
14	NSSF	14,000	1	16	760	2	340	400	500
15	NSSF	14,000	1	16	770	2	350	405	510
16	NSSF	14,000	2	16	770	2	350	405	510
17	NSSF	14,000	2	16	775	2.5	365	400	510
18	NSSF	14,000	1	16	775	2.5	365	400	510

*NF - No Fire

NSSF - Non-Self-Sustaining Fire

SSF - Self-Sustaining Fire

The oil and air temperatures were then increased gradually to 458°K (365°F) and 685°K (775°F) respectively. With mixture temperature ranging from 522 to 560°F (480°F to 550°F) non-self-sustaining fires were ignited in all ten activations. These test results thus showed good correlation with the flammability limits. It should also be noted 9 fires out of 10 attempts were started with ignitor 1 and only 4 out of 8 at ignitor 2 even though the measured mixture temperatures were essentially the same.

Series 4, Run 2

The second test run was performed with an increased air flow rate, 46 stdm³/hr. (27 scfm), and approximately 55°K (100°F) higher air temperature, see Table 12. The oil inlet temperature range was maintained essentially the same as in the prior run. The extreme values of the test variable were:

Hot air flow rate through sump	- 46 stdm ³ /hr. (27 scfm)
Hot air temperature entering hot air chamber	- 716-733°K (820-860°F)
Oil inlet temperature	- 433-444°K (320-340°F)
Oil flow rate	- 0.23-0.57 m ³ /hr. (1-2.5 gpm)
Bearing outer ring temperature	- 469-489°K (385-420°F)
Temperature at ignitor (mixture temperature)	- 544-599°K (520-620°F)

In all nine ignitor actuations, the mixture temperatures were within the flammability limits and in all case fires were ignited. Four of the fires were self-sustaining. Thus again, as in the prior run, excellent correlation existed with the flammability limits.

Series 4, Run 3

The third test run was performed with essentially the same high oil inlet temperature range, a reduced air flow rate and low temperature range. The extreme values of the test variables were:

Hot air flow rate through sump	- 27 stdm ³ /hr. (16 scfm)
Hot air temperature entering hot air chamber	- 622-668°K (660-745°F)
Oil inlet temperature	- 428-450°K (310-350°F)
Oil flow rate	- 0.23-0.57 m ³ /hr. (1-2.5 gpm)
Bearing outer ring temperature	- 462-480°K (375-405°F)
Temperature at ignitor (mixture temperature)	- 516-555°K (470-540°F)

TABLE 12
TEST DATA
SERIES 4 RUN-2

Ignition Attempt (No.)	Results	Shaft Speed (rpm)	Spark Ignitor No.	Air Flow Rate (scfm)	Air In Temp. (°F)	Oil Flow Rate (gpm)	Oil In Temp. (°F)	Brg. Temp. (°F)	Mixture Temp. At Ignitor (°F)
1	*SSF	14,000	1	27	820	1.5	340	410	580
2	NSSF	14,000	2	27	830	1.5	350	410	540
3	SSF	14,000	2	27	840	1.5	355	415	540
4	NSSF	14,000	2	27	845	2	360	415	550
5	NSSF	14,000	1	27	845	2	360	415	560
6	SSF	14,000	1	27	855	1	320	420	620
7	SSF	14,000	2	27	855	1	320	420	550
8	NSSF	14,000	2	27	860	2.5	325	385	540
9	NSSF	14,000	1	27	860	2.5	325	385	520

*NF - No Fire

NSSF - Non-Self-Sustaining Fire

SSF - Self-Sustaining Fire

The mixture temperature throughout the test varied from 17°K (30°F) below to 22°K (40°F) above the LL. In 10 of the 25 activation attempts non-self-sustaining fires were ignited, 8 out of 13 attempts with ignitor 1 and 2 out of 12 attempts with ignitor 2. This same pattern was also noted in run 1. Although the correlation with the flammability limits was not as good as in the previous two runs, it was reasonably good considering that the measured mixture temperatures were only slightly above or below the LL. Reviewing the data with respect to oil flow rates, see Table 13, showed that a higher percentage of fires were ignited with increasing oil flow rates as tabulated in the table below:

<u>Oil Flow Rate</u> <u>m³/hr (gpm)</u>	<u>Ignition</u> <u>Attempts</u>	<u>Fires</u> <u>Ignited</u>
0.23(1)	6	0
0.34(1.5)	7	2
0.46(2)	6	3
0.57(2.5)	6	5

This was also true in run 1.

It is also noted from the test data that the only two fires started with ignitor 2 were with the highest oil flow rate. Therefore, the results suggest that when fires were not ignited it was a result of a lean mixture. This could have resulted from three possible conditions: (1) insufficient oil in the mixture to produce a flammable mixture even if all the oil was evaporated, (2) the equilibrium oil concentration was not adequate to support combustion due to low temperature, (3) the measured temperature was sufficiently high but an equilibrium condition did not exist (the oil temperature did not reach the measured temperature) thus a lower oil-vapor air mixture than expected existed. Although the data would indicate condition 1 existed (higher percentage of fires with increasing oil flow) it is also possible that with increasing oil flow rates a larger number of smaller oil particles were generated. These particles would absorb heat from the air more rapidly and thus the resident time in the air before a flammable oil concentration exists would be shorter.

The two patterns (increased percentage of fires with increased oil flow rate and higher percentage of fires started with ignitor 1) noted in runs 1 and 3 did not exist in run 2 where fires were ignited in all attempts regardless of oil flow rate or ignitor used. However, in run 2, the inlet air temperature and flow rates were higher which could have produced a different oil and air flow path and heat transfer conditions thus resulting in the difference observed which includes the more severe fires being produced.

TABLE 13
TEST DATA
SERIES 4 RUN 3

Ignition Attempt (No.)	Results	Shaft Speed (rpm)	Spark Ignitor No.	Air Flow Rate (scfm)	Air In Temp. (°F)	Oil Flow Rate (gpm)	Oil In Temp. (°F)	Brg. Temp. (°F)	Mixture Temp. At Ignitor (°F)
1	*NF	14,000	1	16	720	1	310	375	530
2	NF	14,000	2	16	720	1	310	385	490
3	NF	14,000	2	16	710	1.5	310	380	490
4	NSSF	14,000	1	16	710	1.5	310	380	500
5	NSSF	14,000	1	16	710	1.5	320	380	500
6	NSSF	14,000	1	16	705	2	330	380	480
7	NF	14,000	2	16	705	2	335	385	490
8	NSSF	14,000	2	16	705	2.5	340	380	480
9	NSSF	14,000	1	16	705	2.5	340	380	470
10	NF	14,000	1	16	680	1	335	405	540
11	NF	14,000	2	16	680	1	335	405	500
12	NF	14,000	2	16	665	1.5	335	400	500
13	NF	14,000	1	16	665	1.5	335	400	520
14	NSSF	14,000	1	16	660	2	340	395	480
15	NF	14,000	2	16	660	2	345	400	510
16	NF	14,000	2	16	660	2.5	350	385	480
17	NSSF	14,000	1	16	665	2.5	350	390	480
18	NF	14,000	1	16	710	1	345	405	540
19	NF	14,000	2	16	725	1	345	405	510
20	NF	14,000	2	16	530	1.5	345	405	510
21	NF	14,000	1	16	730	1.5	345	405	540
22	NSSF	14,000	1	16	735	2	350	400	500
23	NF	14,000	2	16	735	2	350	400	510
24	NSSF	14,000	2	16	745	2.5	350	390	500
25	NSSF	14,000	1	16	745	2.5	350	390	490

*NF - No Fire

NSSF - Non-Self-Sustaining Fire

SSF - Self-Sustaining Fire

A general review of the testing results with respect to the correlation between flammability limits and fire ignitions show that in Test Series 1, 2 and 4 that a reasonable correlation existed. However, in Test Series 3 very poor correlation existed. The major difference existing between Series 3 and the other series was the inlet oil temperature was lower in Series 3. In general, the oil inlet temperature in Series 3 was below 432°K (320°F) when a fire would be expected, but did not occur. In all the other series, the oil inlet temperature was above 432°K (320°F), except for Series 1, Run 4. Thus, the correlation observed would suggest that the comparison of the measured mixture temperature with the flammability limits is a reasonably good criterion to judge the flammability condition within a bearing sump when the oil inlet temperature is above 432°K (320°F).

5.5 Computerized Statistical Evaluation of Test Data

In an effort to obtain additional information and trends from the test data, a computerized statistical evaluation of the test data was performed.

A data base containing the values of the fixed and response variables recorded during the testing program was prepared. The BMDP system of statistical programs developed at the Health Sciences Computing Facility at UCLA was used to interrogate this data base in an attempt to determine additional information and how the occurrence or non-occurrence of fires is affected by these variables. A copy of the form on which the data was transcribed and from which it was keypunched is presented in Figure 14.

Using program BMDP6D12, bivariate scatter plots using three different symbols to denote the "fire condition" (A for no fire, B for non-self-sustaining fire, and C for self-sustaining fire) were initially obtained as combinations of the following choices of X and Y variables; X axis being the controlled variable and the Y axis the dependent variable.

X-Axis

Air Inlet Temperature
Air Flow Rate
Oil Inlet Temperature
Oil Flow Rate

Y-Axis

Oil Outlet Temperature
Outer Ring Temperature
Mixture Temperature

These plots were examined to see if the data points for each fire condition exhibited any clustering and to determine if any correlation existed between the two variables used. A typical bivariate plot is presented in Figure 15.

FIGURE 14

S U M P F I R E D A T A F O R M

FOR STATISTICAL EVALUATION

Card
Col.

Test No.

(If a single digit, put in col. 2)

Run - Attempt No.

(e.g. for attempt 7 in Run 5 write 507)

I. FIXED VARIABLES

Hot Air Inlet Temp (°F)

Where applicable the location of the decimal point is indicated by a "dot". The decimal is not keypunched.

Hot Air Flow Rate (scfm)

Oil Inlet Temp (°F)

Oil Inlet Flow Rate (gpm)

Shaft Speed (1000 RPM)

Ignitor Location

1) Location #1

2) Location #2

Baffle Type

1) Type 1

2) Type 2

II. MEASURED RESPONSES

Oil Outlet Temperature (°F)
(use 999 if missing)

Outer Ring Temperature (°F)

Mixture Temp at Ignitor (°F)
(use 999 is missing)

Fire Condition

0) No Fire 1) Non Self-Sustaining Fire
2) Self-Sustaining Fire

1-2

3-5

6-9

10-11

12-14

15-16

17-19

20

21

22-24

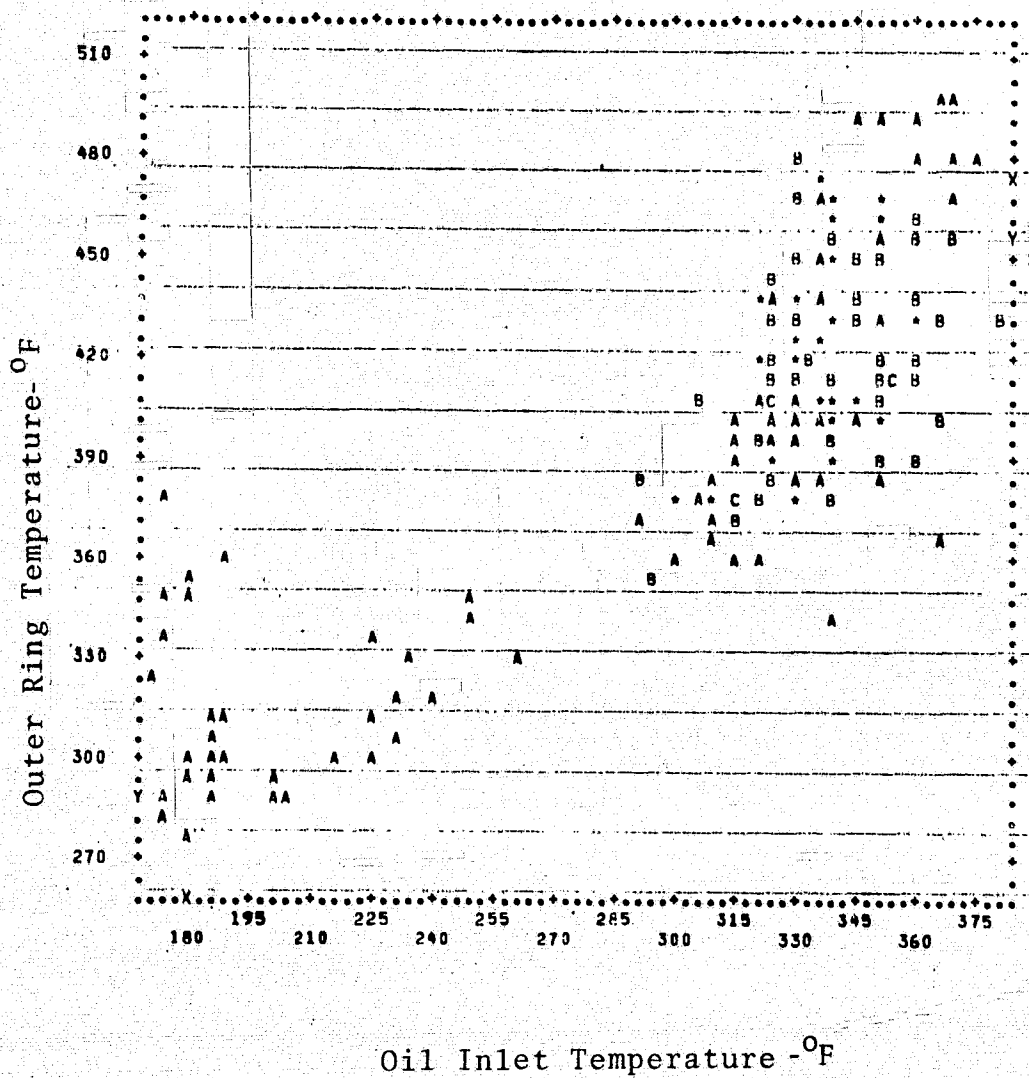
25-27

28-30

31

FIGURE 15

Typical Bivariate Plot



- A - No Fire
- B - Non-Self-Sustaining Fire
- C - Self-Sustaining Fire
- * Multiple Point

The results of these runs, and several other runs using combinations of the controlled variables and calculated variables, essentially verified relationships previously observed. The three plots of the air inlet temperature versus the dependent variables indicated that there was no air inlet temperature used with which fires were not ignited. The major segregations were observed when either oil inlet temperature, oil outlet temperature, or bearing temperature were one of the variables used in the plot. No fires were ignited when the oil inlet temperature was below 432°K (290°F), or the oil out temperature was below 450°K (350°F). Only one fire was ignited when the bearing outer ring temperature was below 461°K (370°F), see Figure 16. This could be expected as the relationship of oil inlet temperature and fire ignitions had been previously noted and the oil out and bearing temperatures are a direct function of the oil in temperature. A plot of oil inlet versus oil out temperature resulted in a correlation coefficient of 0.98.

An examination of the plots incorporating mixture temperatures indicated that it was more directly related to the air inlet temperature than the oil inlet temperature. The plot of mixture temperature versus air inlet temperature had a correlation coefficient of 0.77 while the correlation coefficient between the mixture temperature and oil temperature was only 0.33. This would indicate, as previously noted, that the oil was not always mixing with the air before it passed the ignitor location and the temperature measured.

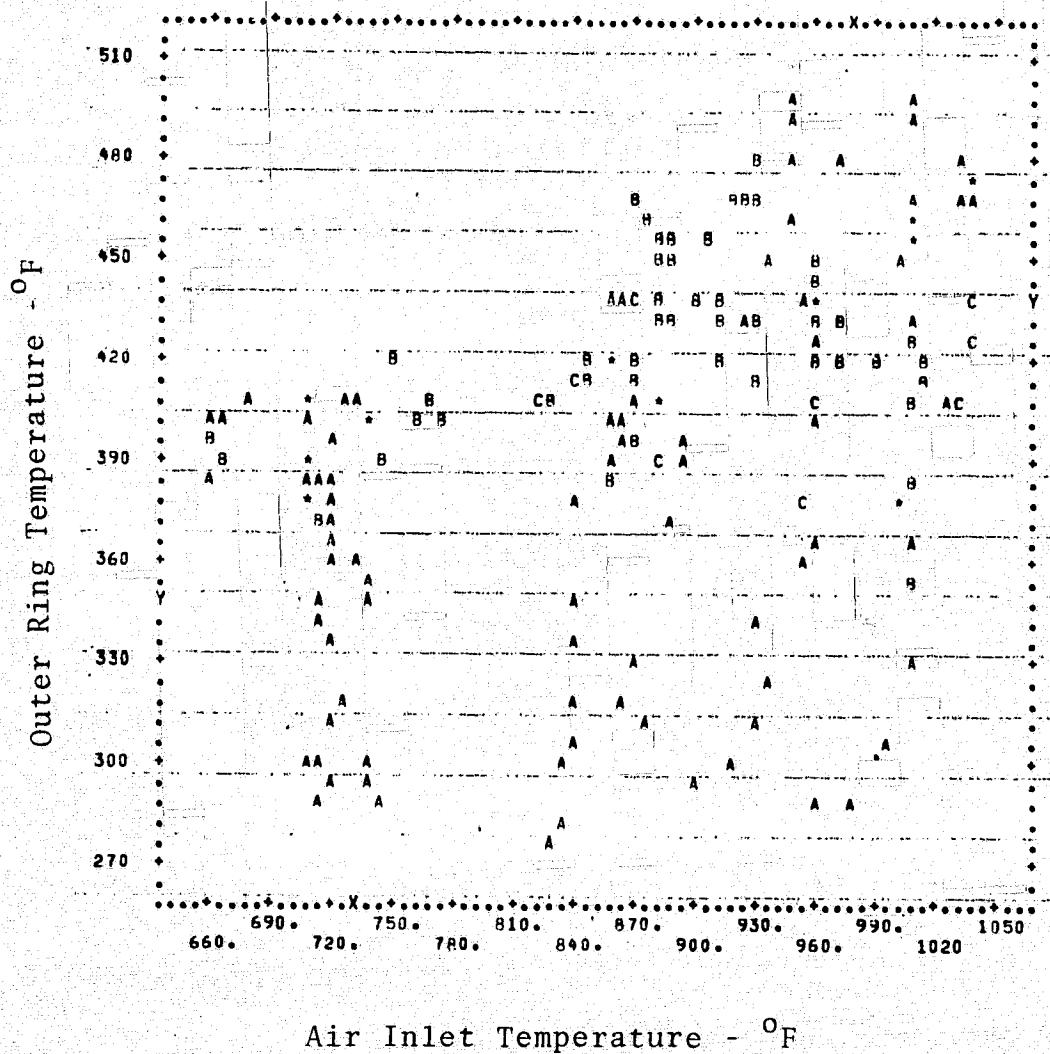
The bivariate plot of the mixture temperature versus oil inlet temperature, presented in Figure 17, shows the reasonably good correlation between the mixture temperature and the flammability limits when the oil inlet temperature was above 432°K (320°F) as had been previously noted.

The basic problem in evaluating the test data from the bivariate plots is that only two variables can be evaluated at a time and no relationship of a particular point with a given test or other variable could be established to provide further understanding of its particular location. Even with this drawback, the plots generally substantiated the prior observations.

Further statistical analyses were performed using Program BMDP7M. This program performs stepwise linear discriminate function analysis. It seeks to form a linear combination of variables which best classifies a case into groups such as fire or no fire. It introduces the variables sequentially (stepwise) choosing at each step the variable that best contributes to the ability to correctly classify cases. For the sump fire data, the variables

FIGURE 16

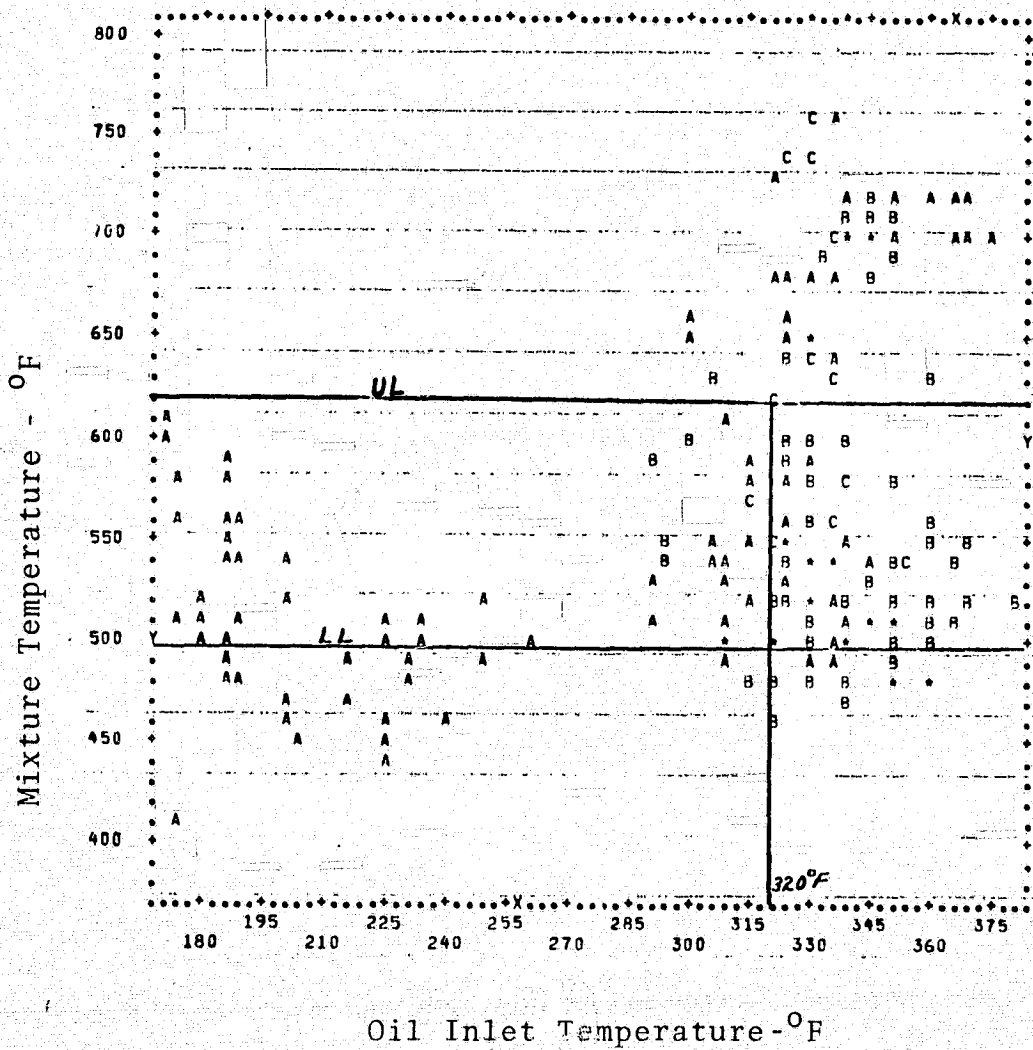
Bivariate Plot - Outer Ring Vs. Air Inlet Temperatures



- A - No Fire
- B - Non-Self-Sustaining Fire
- C - Self-Sustaining Fire
- * - Multiple Point

FIGURE 17

Bivariate Plot - Mixture Vs. Oil Temperatures



- A - No Fire
- B - Non-Self-Sustaining Fire
- C - Self-Sustaining Fire
- * Multiple Point

selected by the program were: (1) oil outlet temperature, (2) oil inlet temperature, and (3) oil flow rate. The classification matrix was found to be:

Group	<u>Number of Cases Classified into Group</u>	
	No Fire	Fire
No Fire	49	15
Fire	8	30

The total percentage of correct classifications was 77%. The basic significance of this evaluation was the indicated relative importance of the oil temperature and flow rate.

6.1 Description of Analytical Study

The results of the test program, as presented and discussed in Section 5, definitely indicate that the flammability condition within a jet engine bearing sump are not homogeneous and vary appreciably with changes in the input variables even in a given sump configuration. Even though general guidelines can be established from the test results which are considered applicable for reducing the probability of fires in any sump, differences in sump configurations compound the problem of determining the flammable conditions present. Therefore, to take the initial step in obtaining an approach which would permit the prediction of the flammability condition with a sump, an analytical study of a simplified configuration was performed.

The purpose for this study was to formulate mathematical models which would:

1. Simulate the generation of oil vapor in a moving oil droplet - air mixture.
2. Evaluate the combustion hazard of a given two phase mixture subjected to a specified thermal input.

Two phase characterization at any flow location would be performed. The determination of the associated residence time for a critical droplet size distribution would be computed given knowledge of initial velocity conditions. Thus, for a given thermal stimulus of any condition characterized at a specific location, computations would be made to see if a flame would propagate, burn locally or extinguish itself.

The mathematical models were formulated so that the influence of physical properties, geometry, temperature, velocity and size distribution of droplets etc. could be observed. The documentation for their effort is attached in Appendix I. Sample computations are documented for n-decane due to the ready availability of pertinent physical characterization of the material.

6.2 Computerized Analysis

The computer program as previously described and presented in Appendix I was exercised on this program for two specific purposes: (1) the initial 13 runs were a parametric study to determine the affect on the generation of oil vapor due to changes in specific independent variables and (2) the second set of 12 runs were performed to determine if results would correlate with

test data. Therefore, independent variables similar to those in specific tests were selected. In addition the geometric variables, e.g., flow area and perimeter, were established as representative of the test rig configuration when it was envisioned as representative of the test rig configuration when it was envisioned as a straight tube with the flow moving axially (similar to geometry used in analytical program).

The air properties used in the computer analysis were taken from well established values presented in any thermodynamics text book. The oil properties, however, were much more difficult to obtain due to its complexity and the infrequency that its thermodynamic properties are required. The oil property values used in the computer program and presented below were obtained from literature, various oil companies, and when necessary theoretically established by calculation. Thus these values must be considered as representative values and not firm values.

List of Oil Property Values Used in Computer Program

CPFU - Specific Heat of Gaseous Fuel - $\text{Btu/lb}_m^\circ\text{F}$ - 0.3

CPL - Specific Heat of Liquid Fuel - $\text{Btu/lb}_m^\circ\text{F}$ - 0.536 at 100°F
 - 0.561 at 300°F
 - 0.450 at 450°F

HFUO - Enthalpy of Formation of Gaseous Fuel - Btu/lb_m - 6330 at 300°F
 - 6375 at 400°F

HLO - Enthalpy of Formation of Liquid Fuel - Btu/lb_m - 6250

Note: Latent Heat of Vaporization = HFUO-HLO

WFU - Molecular Weight of Fuel - $\text{lb}_m/\text{lb}_{\text{mol}}$ - 556

PF1, PF2 - Vapor Pressure Corresponding to TL_1 , TL_2 -psia -
 0.0067, 0.02

TL_1 , TL_2 - Temperature Corresponding to PF1, PF2 - $^\circ\text{F}$ - 425, 475

RHOL - Density of Liquid Fuel - lb_m/ft^3 - 53.7

For the parametric study a baseline set of values were selected for each variable and in turn changed to two other values, one lower and one higher, while maintaining all other variables at the baseline value. The baseline values were selected to represent

a mean and the other two values the upper and lower limits that may be expected in a sump surrounding an engine bearing. The six variables evaluated and the values used in the study are presented in Table 14. Case number 20 is the baseline value. The case numbers noting the changes in each variable are as follows:

<u>Case No.</u>	<u>Variable Changed</u>
21, 22	Air Inlet Temperature
31, 32	Air Flow Rate
41, 42	Oil Inlet Temperature
51, 52	Oil Flow Rate
61, 62	Oil Droplet Diameter
71, 72	Oil Velocity Relative to Air Velocity

A graph showing how the gas (air and vapor) temperature and the vapor mass fraction of the gas change with axial displacement in the tube is presented in Figure 18 for cases 20 and 22 where the only change is the inlet air temperature. The graph shows that an increase in gas temperature from 600°F to 800°F more than doubles the vapor concentration. Graphs showing the effect of the changes made in each variable on the volume ratio of oil vapor to air are presented in Figures 19-24. Each graph consists of three lines which represent the percentage of oil vapors after a flow distance of 0.3, 0.5 and 1 ft. The upper and lower flammability limits, expressed in percent vapor, are marked on the graphs to provide a relative evaluation of the significance of the vapor concentration with respect to the variable change and the flow distance of the oil in the air stream. The computer printout sheets for the input parameters and the computed values at the first five positions calculated for Case 20 are presented in Tables 15 and 16 respectively.

These graphs show that the vapor concentrations change as expected with the variable changes, i.e. the concentration increases with increases in air inlet temperature, oil inlet temperature and oil flow rate, and decrease with increases in air flow rate, droplet size, and the relative velocity of the oil to air. The variable changes shown to have the greatest influence on the vapor concentration are air inlet temperature and oil droplet size.

The effect of changes in oil inlet temperature and flow rate, which the test data indicated had a major influence on flammability, are shown to have lesser effect than expected. This would indicate that the greater numbers of fires resulting

TABLE 14

SETS OF VARIABLE VALUES USED IN PERFORMING THE
PARAMETRIC STUDY

<u>Variables</u>	Air Inlet Temp.	Air Flow **	Oil Inlet Temp.	Oil Flow **	Oil Droplet Diameter	Oil Velocity Relative To Air Velocity
Case No.	(°F)	(scfm)	(°F)	(cfm)	(inches)	(oil/air)
* 20	600	8	270	0.014	0.020	0.5
21	400	8	270	0.014	0.020	0.5
22	800	8	270	0.014	0.020	0.5
31	600	14	270	0.014	0.020	0.5
32	600	4	270	0.014	0.020	0.5
41	600	8	170	0.014	0.020	0.5
42	600	8	370	0.014	0.020	0.5
51	600	8	270	0.008	0.020	0.5
52	600	8	270	0.020	0.020	0.5
61	600	8	270	0.014	0.004	0.5
62	600	8	270	0.014	0.040	0.5
71	600	8	270	0.014	0.020	0.25
72	600	8	270	0.014	0.020	1.0

* Row 20 represents baseline values.

** Values are not total flow rates used in test program, but modified to be representative in simplified physical model.

FIGURE 18

Graph of Calculated Change in Gas Temperature
and Vapor Mass Fraction With Distance Traveled

(Cases 20 & 22)

96

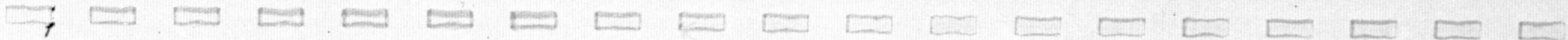
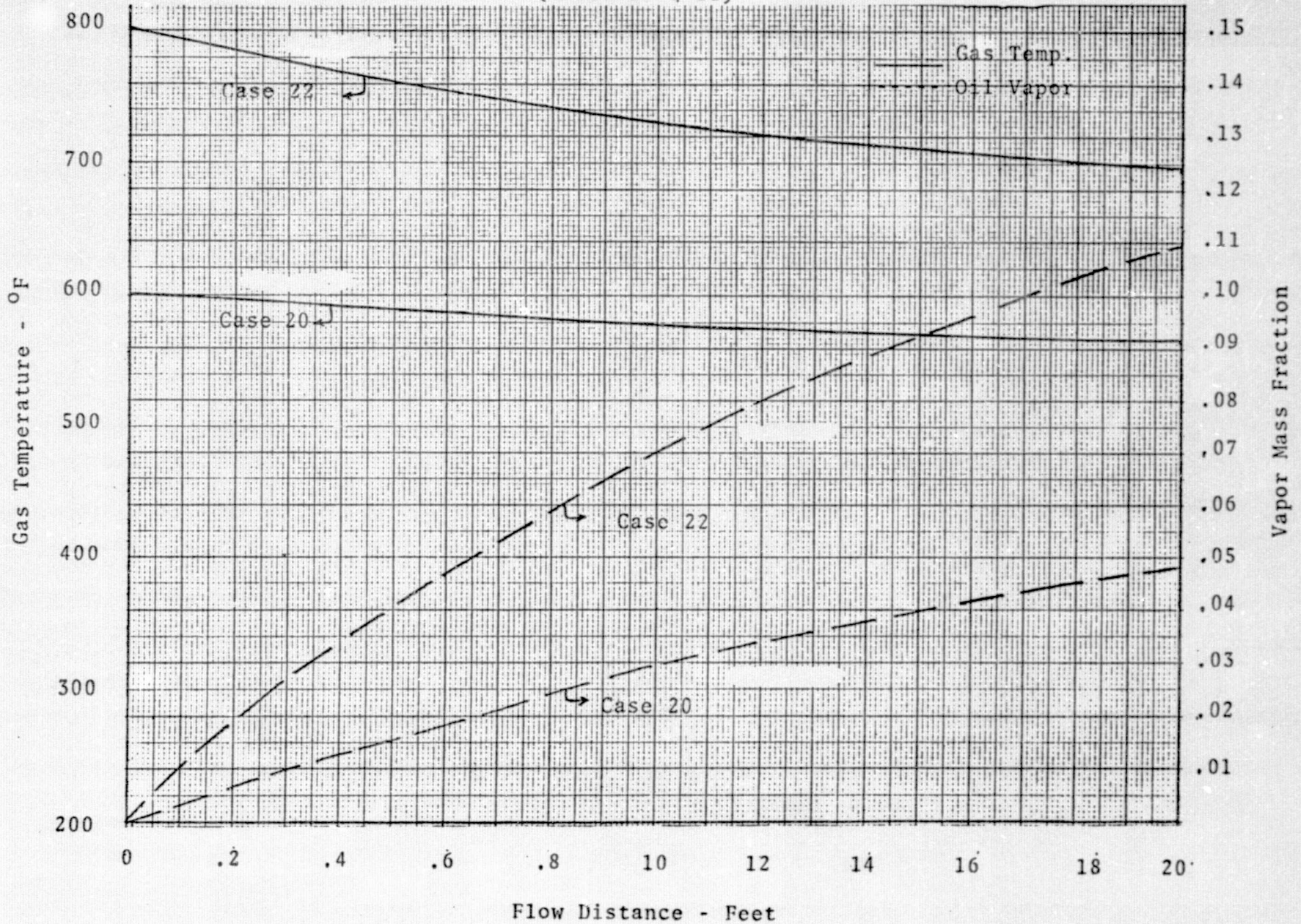


FIGURE 19

Air Temperature Effect on Oil Vapor Generation

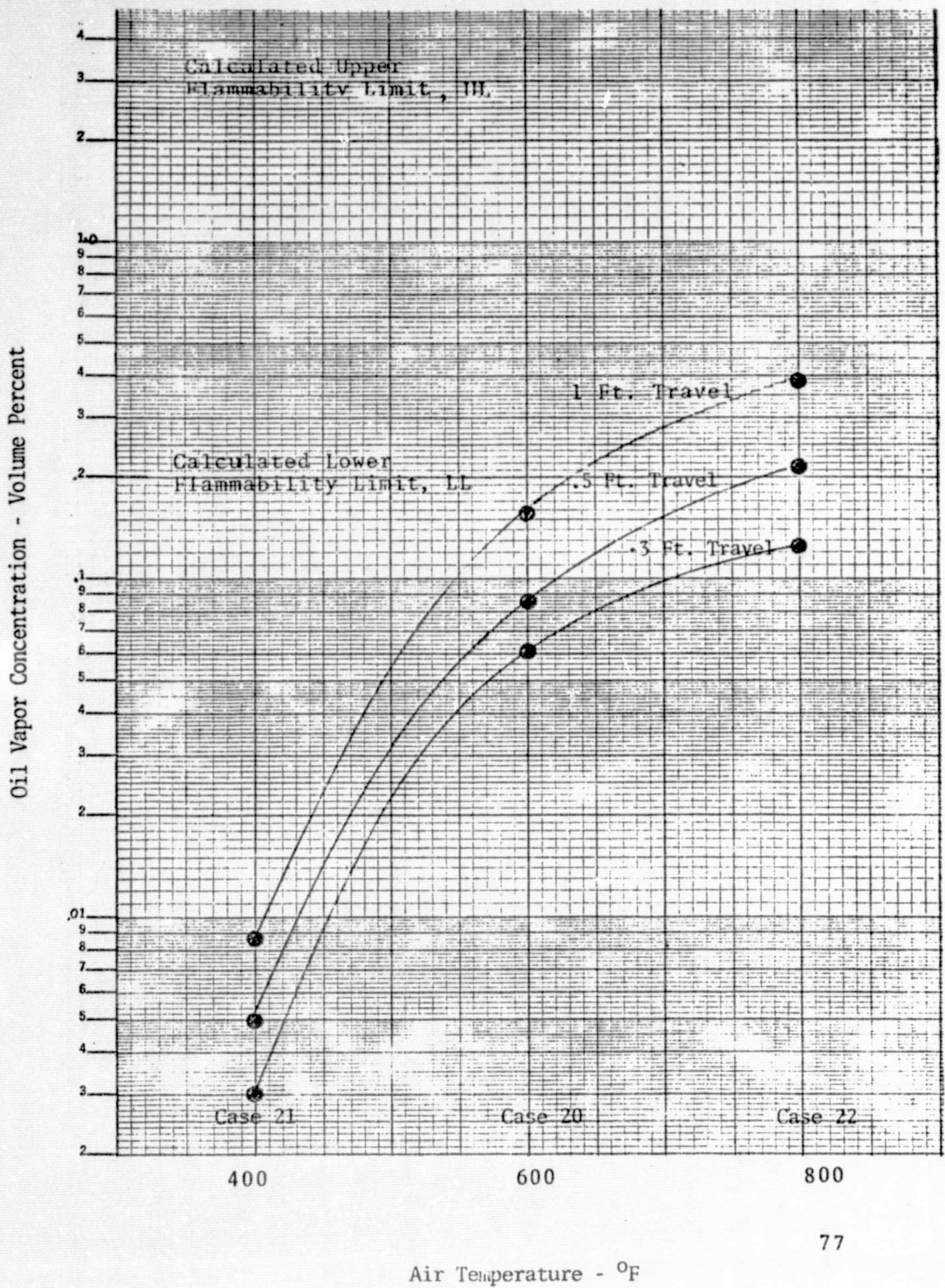
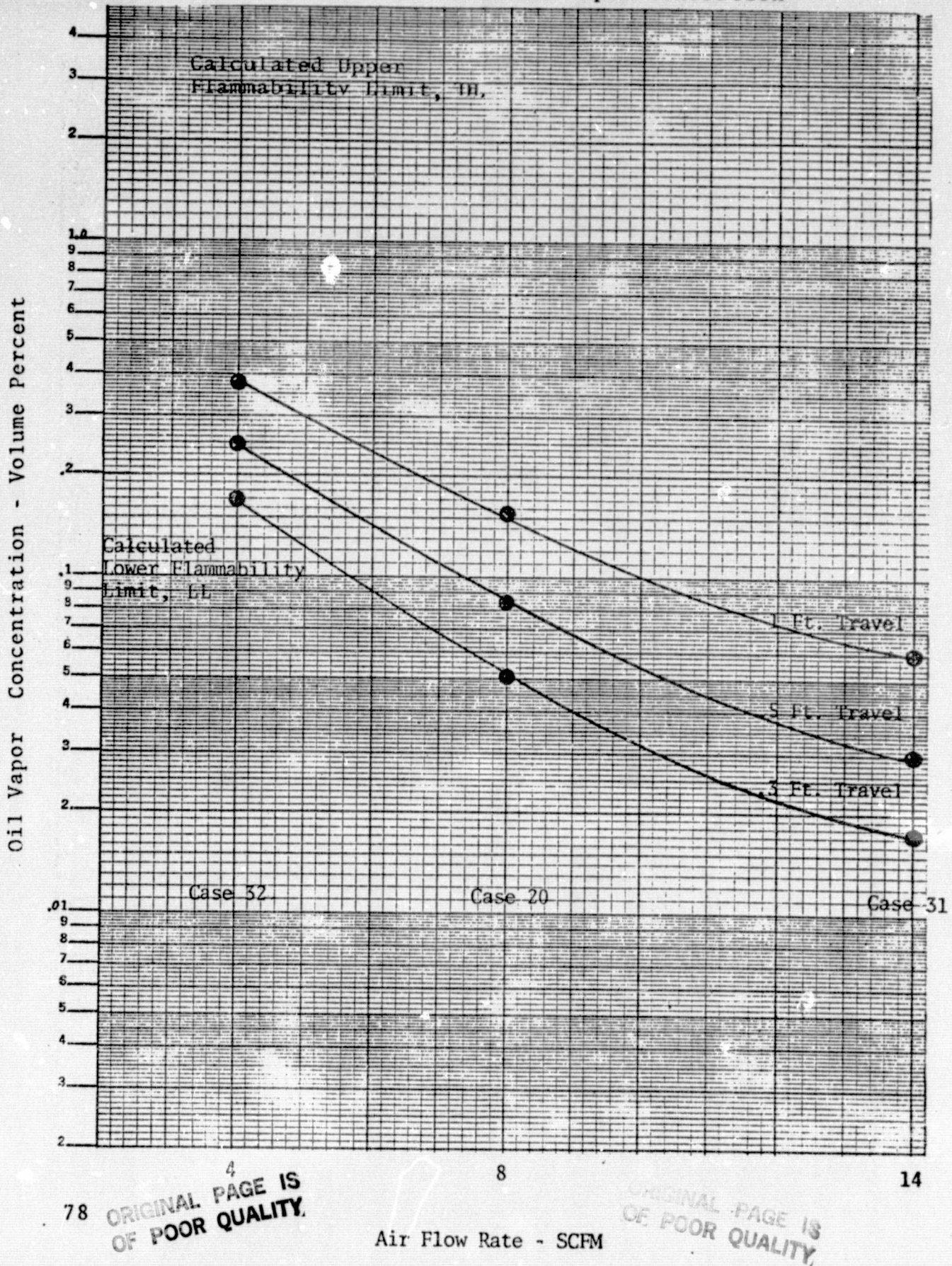


FIGURE 20

Air Flow Rate Effect On Oil Vapor Generation



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FIGURE 21

Oil Inlet Temperature Effect on Oil Vapor Concentration

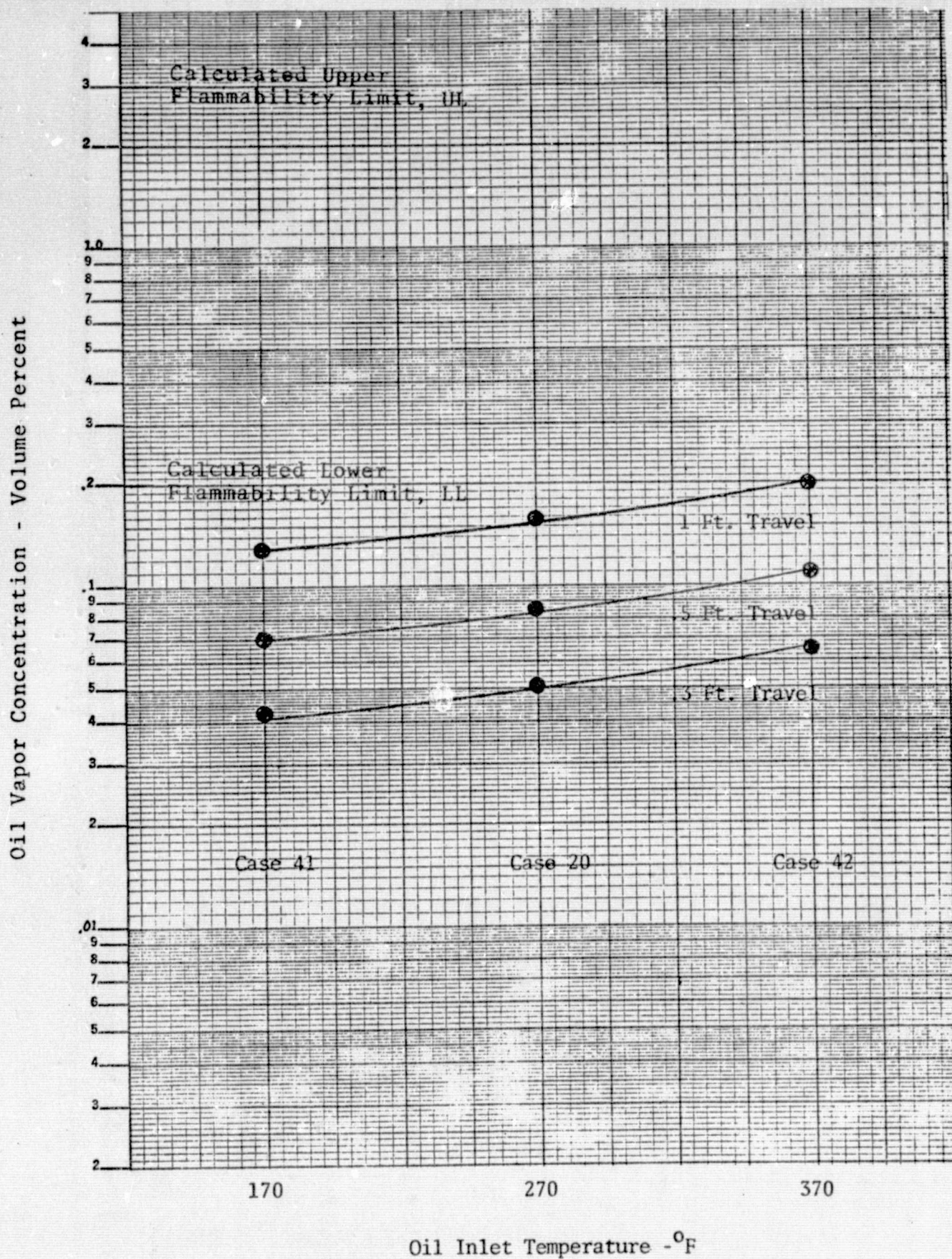


FIGURE 22

Oil Inlet Flow Rate Effect On Oil Vapor Concentration

Oil Vapor Concentration - Volume Percent

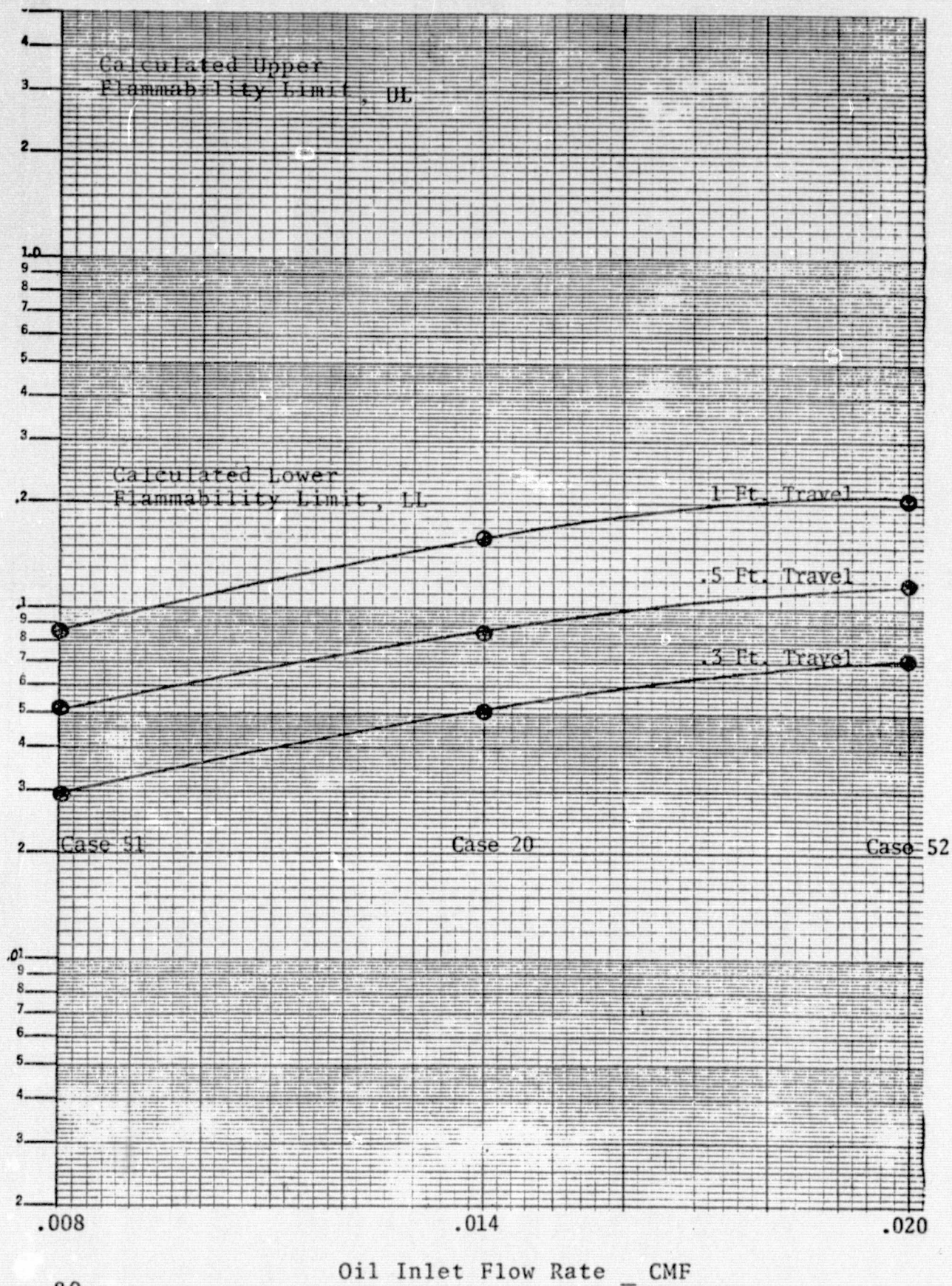


FIGURE 23

Oil Droplet Diameter Effect On Oil Vapor Concentration

Oil Vapor Concentration - Volume Percent

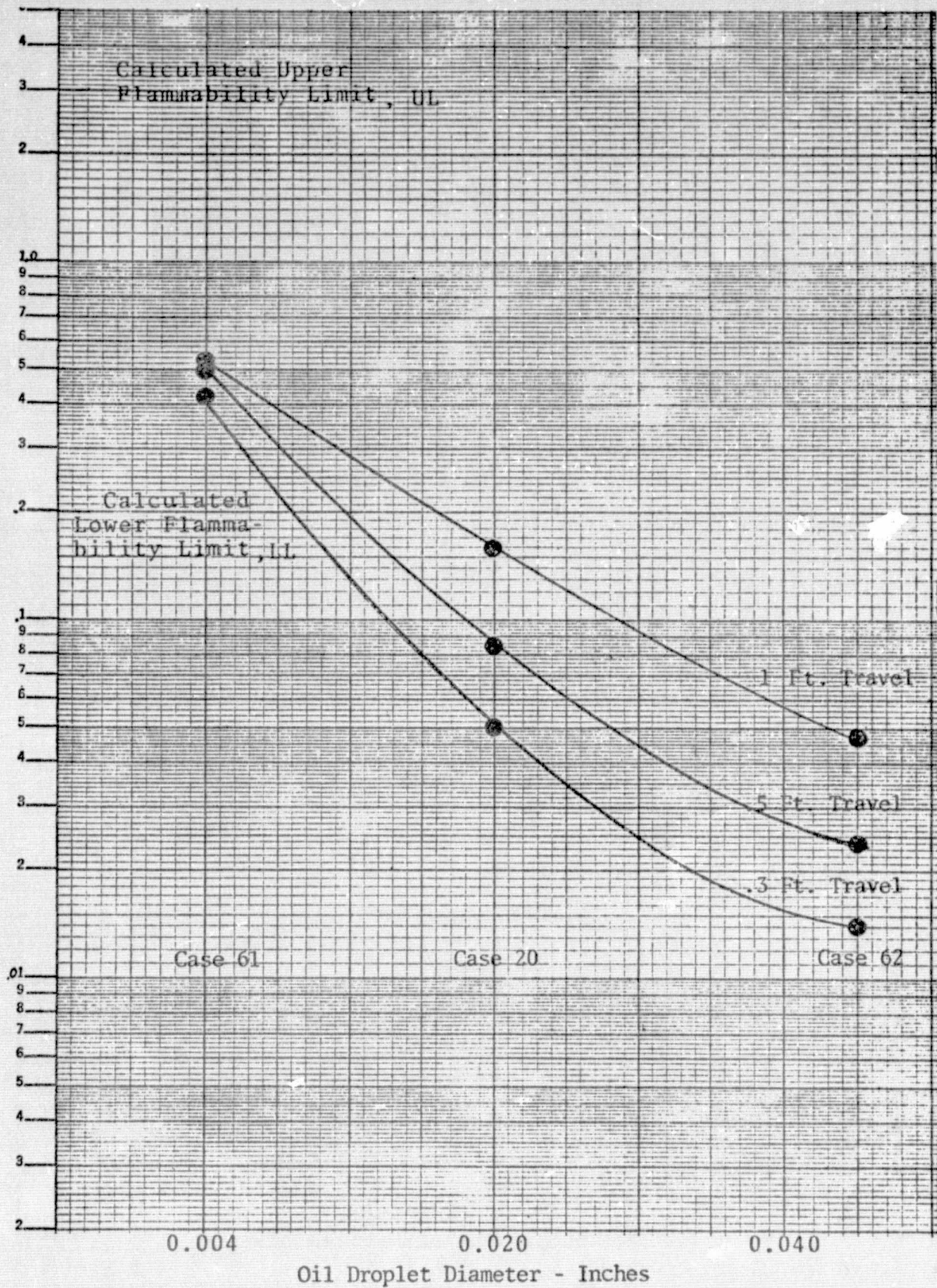


FIGURE 24

Effect of Oil Droplet Velocity Relative to Air Velocity On
Oil Vapor Concentration

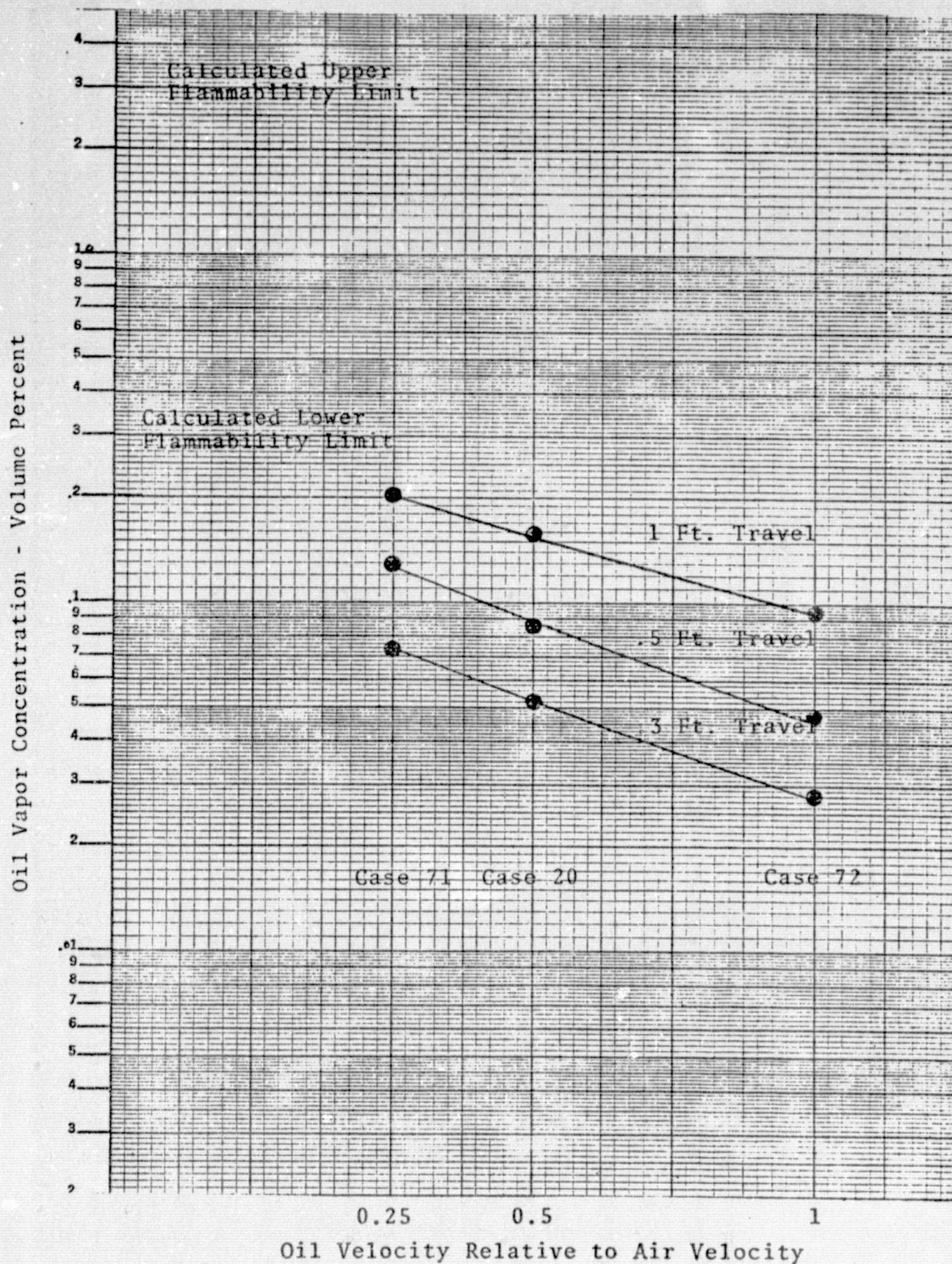


TABLE 15

COMPUTER PRINTOUT OF AIR AND OIL PROPERTIES, AND
SYSTEM PARAMETERS FOR CASE 20

(Steady Flow Droplet Evaporation Program)

AIR PROPERTIES

CPA,BTU/LBM=	.24460+00
TAIR,DEGF=	.60000+03
WAIR,LBM/LBMOLE=	.28950+02
COND,BTU/HR/FT/DEGF=	.22500-01
XMUG,LBM/FT/SEC=	.17500-04

FUEL PROPERTIES

CPEU,BTU/LBM/DEGF=	.30000+00
CPL,BTU/LBM/DEGF=	.60000+00
HFUD,BTU/LBM=	-.62500+04
HLO,BTU/LBM=	-.63760+04
WFU,LBM/LBMOLE=	.55600+03
TL,DEGF=	.27000+03
RHOL,LBM/FT**3=	.53700+02

DROPLET PROPERTIES

NGROUP=	1
RO(1),MICRONS=	.25000+03
FP(1)=	.10000+01
VO(1),FT/SEC=	.55730+01
PF1,PSIA=	.10000+00
TL1,DEGF=	.55000+03
PF2,PSIA=	.23000-02
TL2,DEGF=	.37500+03
PRES,PSIA=	.14700+02

SYSTEM PARAMETERS

VOLA,SCFM=	.80000+01
VOLS,CFM=	.14000-01
AREA,FT**2=	.24000-01
HTCOEF,BTU/HR/FT**2/DEGF=	.00000
PERIM,FT=	.62500+00
NWALL=	2

TABLE 16

COMPUTED VALUE FOR FIRST FIVE LOCATIONS - CASE 20
(Steady Flow Droplet Evaporation Program)

84

	POSITION, FT=	.00000			
	YAIR, LBM AIR/LBM GAS=	.10000+01			
	YFU, LBM GASEOUS FUEL/LBM GAS=	.00000			
	RHO, TOTAL LBM/FT**3=	.13103+00			
	TGAS, DEGF=	.60000+03			
	TWALL, DEGF=	.39000+03			
	TSURF, DEGF=	.27000+03			
	GAS VELOCITY, FT/SEC=	.11155+02			
R(1), MICRONS=	.25000+03	YS(1), LBM SPRAY/LBM=	.55615+00	VELOC=	.55730+01
	POSITION, FT=	.10000+00			
	YAIR, LBM AIR/LBM GAS=	.99786+00			
	YFU, LBM GASEOUS FUEL/LBM GAS=	.21435-02			
	RHO, TOTAL LBM/FT**3=	.12761+00			
	TGAS, DEGF=	.59854+03			
	TWALL, DEGF=	.39000+03			
	TSURF, DEGF=	.54077+03			
	GAS VELOCITY, FT/SEC=	.11140+02			
R(1), MICRONS=	.24986+03	YS(1), LBM SPRAY/LBM=	.55520+00	VELOC=	.57828+01
	POSITION, FT=	.20000+00			
	YAIR, LBM AIR/LBM GAS=	.99385+00			
	YFU, LBM GASEOUS FUEL/LBM GAS=	.61548-02			
	RHO, TOTAL LBM/FT**3=	.12464+00			
	TGAS, DEGF=	.59580+03			
	TWALL, DEGF=	.39000+03			
	TSURF, DEGF=	.54068+03			
	GAS VELOCITY, FT/SEC=	.11113+02			
R(1), MICRONS=	.24959+03	YS(1), LBM SPRAY/LBM=	.55340+00	VELOC=	.59770+01
	POSITION, FT=	.30000+00			
	YAIR, LBM AIR/LBM GAS=	.99016+00			
	YFU, LBM GASEOUS FUEL/LBM GAS=	.98400-02			
	RHO, TOTAL LBM/FT**3=	.12206+00			
	TGAS, DEGF=	.59329+03			
	TWALL, DEGF=	.39000+03			
	TSURF, DEGF=	.54057+03			
	GAS VELOCITY, FT/SEC=	.11088+02			
R(1), MICRONS=	.24934+03	YS(1), LBM SPRAY/LBM=	.55174+00	VELOC=	.61578+01
	POSITION, FT=	.40000+00			
	YAIR, LBM AIR/LBM GAS=	.98676+00			
	YFU, LBM GASEOUS FUEL/LBM GAS=	.13244-01			
	RHO, TOTAL LBM/FT**3=	.11980+00			
	TGAS, DEGF=	.59098+03			
	TWALL, DEGF=	.39000+03			
	TSURF, DEGF=	.54047+03			

from the higher oil flow rate was not totally the result of increased oil in the air; but suggest that increased oil flow also resulted in smaller oil droplet sizes. This is certainly possible since the increased oil velocity through the nozzles could produce some atomization and the greater impingement velocity on the bearing could also produce smaller particles. The possibility of these two conditions emphasizes the importance of a properly designed lubricant nozzle system and suggest an area of investigation.

The increase in the number of fires with an increase in oil inlet temperature, which was very pronounced in the testing, could also have resulted partially from an increase in small particles. The size of a droplet formed is directly proportional to surface tension which varies inversely with temperature. Therefore, at higher temperatures, smaller particles would be formed.

Several other significant points noted from the graphs are:

- (1) The oil vapor concentration are within the flammability limits in several cases.
- (2) The change in the variable investigated is significant, i.e. the resulting change in the vapor concentration is often changed from an inflammable mixture to a flammable mixture.
- (3) With oil particle sizes of approximately 0.004 inches in diameter, significant vapor is produced in a very short residence time in the air stream. Thus, further suggesting the importance of maintaining as large of oil particle sizes as possible within the sump.
- (4) In none of the cases examined was there a vapor concentration above the calculated upper flammability limit.

Additional analyses were performed to check if correlation existed between the analytical and test results. Variable values representative of specific tests were selected. The tests selected for evaluation were: (1) Series 2, Run 2 where fires were ignited in some actuations, (2) Series 3, Run 1 where no fires were ignited and (3) Series 4, Run 2 where fires were ignited in all attempts. Changes to the values of oil and air flow rates were made to minimize the geometrical differences between the test rig and cylindrical tube use for math modelling. For each selected test run, the air and oil temperature and air

flow rates were considered constant while two oil droplet sizes and flow rates were incorporated for evaluation. The variable values used in the resulting twelve test runs are presented in Table 17.

The resulting oil vapor concentrations at both 0.3 ft. and 0.5 ft. travel are also represented. These two lengths are considered to be representative of the average flow path to an ignitor location. By comparing the calculated concentrations with the flammability limits, the computed values are quite reasonable. In computer runs 1-4, representative of Test Series 2, Run 2 where some fires were ignited, the analysis indicated that fires would ignite with the higher oil flow rate and smaller droplet sizes. In computer runs 5-8, representative of Test Series 3, Run 1 where no fires were ignited, the analysis also indicates that no fires would be expected. In computer runs 9-12, representative of Test Series 4, Run 2 where fires were ignited in all attempts, the analysis indicated that the concentration was too lean to burn and fires would not have been expected. The agreement between test and computed data was actually better than expected considering the assumptions that had to be made in attempting to account for the differences in the configuration of the rig and that used in formulating the math model. In addition the oil droplet size and the quantity of oil entering the air stream had to be established by engineering judgement.

In general, the analysis performed with the two phase flow (liquid and vapor) program indicated that the mathematical model and the resulting computerized program is a feasible method for determining the oil vapor concentration in a two phase flow through a cylindrical tube. Although the checks performed with test data indicated that the results were logical, further comparison with test data obtained in a cylindrical test rig should be performed. The testing should be performed with variable oil droplet sizes, temperature, and flow rates and variable air temperature and flow rates. In addition, work should be performed to evaluate the size of oil droplets generated by an oil stream impinging on a rotating bearing. This work should include the effect of changes in oil flow rate and temperature, bearing speed, nozzle design and stream diameter, and the direction and location of the contact. Such values will be necessary in future analytical studies and could provide information on decreasing the incidence of sump fires by controlling the injection of the bearing lubricant.

TABLE 17

SETS OF VARIABLE VALUES USED TO CHECK CORRELATION WITH TEST RESULTS

Variables Case No.	Air Inlet Temp. (°F)	Air Flow* Rate (scfm)	Oil Inlet Temp. (°F)	Oil Flow* Rate (cfm)	Oil Droplet Diameter (inches)	Oil Velocity Relative To Air Velocity (Oil/Air)	Oil Vapor Concentration (Vol. %)	
							.3 ft.	.5 ft.
1	700	14.5	360	.0083	.008	0.5	.12	.19
2	700	14.5	360	.0167	.008	0.5	.23	.36
3	700	14.5	360	.0083	.080	0.5	.001	.002
4	700	14.5	360	.0167	.080	0.5	.003	.005
5	600	14.5	180	.0083	.008	0.5	.04	.07
6	600	14.5	180	.0167	.008	0.5	.09	.13
7	600	14.5	180	.0083	.080	0.5	.000	.001
8	600	14.5	180	.0167	.080	0.5	.001	.002
9	570	14.5	340	.0083	.008	0.5	.05	.13
10	570	14.5	340	.0167	.008	0.5	.09	.14
11	570	14.5	340	.0083	.080	0.5	.000	.001
12	570	14.4	340	.0167	.080	0.5	.001	.002

*Values are not total flow rates used in test program, but modified to be representative in simplified physical model.

changes within the sump, resulting from input changes, were evaluated, the effects of the input change (oil temperature, oil flow rate, air temperature, air flow rate) was different at each monitored location. Even though the resulting trends from a change were generally the same throughout the sump, the magnitudes would differ. An increase in the oil inlet temperature, air inlet temperature or flow rate resulted in higher mixture temperatures, and increase in oil flow rate resulted in lower mixture temperatures.

Therefore, it was concluded that no particular input parameter, with the possible exception of oil-in temperature, or set of parameter, could be established from which a firm decision, with respect to flammability conditions within the sump, could be established. This condition varies drastically within the sump for a given set of input conditions, i.e. the mixture in one location could be combustible while at another location non-combustible. This results from the complexity of the air flow paths, dispersion of the oil and the oil droplet sizes.

The trends that conditions are more susceptible to fire ignition with increased oil flow and temperature are influenced as much by the generation of smaller oil particles as the greater oil dispersion and lesser heat transfer requirement from the air to generate vapor. This conclusion is based primarily on the results of the analytical study where the oil particle size was shown to have a major influence on the vapor generation rate with respect to residence time in the air.

A reasonably good correlation was found between the mixture temperature at the spark-ignitor and the calculated flammability limits expressed as temperature. This evaluation technique provides a comparatively easy way of judging the combustible condition within a bearing sump. If the mixture temperature is between the flammability limits the possibility of a fire starting in the presence of an ignition source must be considered more probable than if it is outside the limits. This technique, however, is not completely reliable [it was found to be more reliable when the oil inlet temperature was above 432°K (320°F)] as there are two conditions which would alter the expected combustible condition: (1) insufficient oil in the mixture to produce a combustible mixture even if all the oil was evaporated, and (2) the temperature of the oil droplets not being raised to the level of the air temperature. These two conditions indicate the importance of removing the oil from the sump as quickly as possible.

7.0 CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions

From the evaluation of the input parameters as criteria for determining the flammability conditions within the test rig bearing sump several conclusions could be established. Fires could be ignited over the full range of hot air flow rates evaluated, 7 to 49 stdm³/hr. (4 to 29 scfm), and over the complete range of inlet air temperature to the hot air chamber evaluated, 724 to 833°K (845 to 1040°F). The severity of the fires ignited were in general proportional to the hot air flow rate; the greater the flow rate the more severe the fires. With low air flow rates of 7 to 14 stdm³/hr. (4 to 8 scfm) only minor non-self-sustaining fires were ignited. With flow rates of 20 to 27 stdm³/hr. (12 to 16 scfm) the fires were still non-sustaining but much higher mixture temperatures resulted when a fire was ignited. The only self-sustaining fires ignited occurred with air flow rates from 41 to 49 stdm³/hr. (24 to 29 scfm). Thus good seals with high reliability should reduce the probability of fires occurring in the application.

Fires were also ignited over the full range of oil flow rates evaluated, 0.23 to 0.57 m³/hr. (1 to 2.5 gpm). In general, the tendency for fires to be ignited was greater with increases in oil flow rate. This condition most likely existed because there was a greater dispersion of oil droplets with the higher flow rates. None-the-less, self-sustaining fires were ignited with all flow rates evaluated.

This condition would also suggest that the probability of fire would be reduced by decreasing the oil flow rate. However, the oil rate should not be reduced to the point where it would influence the operability of the bearing. These reductions could however also produce high bearing temperatures increasing the generation of oil vapor in the bearing which could negate the purpose of decreasing the flow.

The temperature of the oil supplied to the test bearing for lubrication and cooling was the only input parameter that produced a limiting value with respect to fire ignition. A fire never ignited when the oil inlet temperature was below 417°K (290°F), even when the other input variables were at or near the maximum value of the evaluated range.

Uniform conditions were never generated within the sump with any of the imposed input combinations. The mixture temperatures measured within the sump varied from point to point in all tests. In those tests where detailed examinations of the

The fact that a combustible ratio could exist, due to the lack of thermal equilibrium, when the measured temperature is well above the upper flammability limit should also be noted. Therefore, the upper limit should be considered less reliable than the lower limit. In none of the test cases evaluated was there positive evidence that a fire could not be ignited because the mixture was too rich. This observation was further emphasized by the fact that in none of the cases checked analytically by the computer program was a mixture simulated that was too rich to burn even in a flow distance of one foot.

The statistical analysis techniques used to evaluate the test data were shown to be feasible approaches to establish trends and relative importance of test variables with respect to combustible conditions in the sump. The basic conclusions obtained from these analyses were in agreement with those obtained by direct analysis of the data. A drawback experienced with the bivariate plot approach was that no relationship of a plotted point, representing a particular fire ignition attempt, could be established with any test variable other than the two used in plotting. Thus no further understanding or knowledge could be obtained from its location. The stepwise linear discriminate function analysis further emphasized the importance of the oil temperature and flow rate with respect to flammability.

Two basic mathematical programs were successfully formulated and computerized to aid in the study of the flammability conditions in a jet engine bearing sump. The first program traces two phase flow, liquid and vapor, in a cylindrical geometry. The second program considers the ignition of the vapor in air mixture by an ignition source and allows the determination of combustion or its absence.

The second program was exercised with a two phase mixture of decane which showed proper operation. The first program was successfully utilized in a parametric study to determine the influence on the generation of oil vapor due to changes in specific independent variables. In all cases, vapor concentrations varied as would be expected with the variable change imposed. Air temperature and oil droplet size were shown to have the major effect on the vapor generation rate. In addition, several runs were performed using variable values representative of those imposed in specific test runs to determine what correlation existed between the analytical and experimental results. The correlation was better than expected considering the assumptions that had to be made to compensate for the major differences in geometry of the rig and that used in establishing the mathematical model.

7.2 Recommendations

This program has established directions with respect to change in input variables which should reduce the probability of sump fires and suggest a possible technique of judging flammability condition within a bearing sump. It has also shown that it is highly unlikely that flammability conditions within a sump can be ascertained by evaluating only the selected input parameters. To obtain further knowledge, which will be valuable during a sump design to minimizing the presence of flammable conditions within the sump, a continuation of the analytical approach initiated on this program is recommended.

The recommended program would include both analytical and experimental work. It is suggested that the experimental effort be directed at establishing the flow path and particle size of the oil droplets dispersed from a jet lubricated bearing. In addition, verification testing should be performed to verify the accuracy of the computerized programs developed during this study. This would include the design and manufacture of a simple cylindrical test stand in which the oil and air flow rate, and oil droplet size could be readily established, thus permitting the evaluation of the vapor generation rate and flammability conditions with respect to axial flow distance. Any necessary changes in the developed analytical programs could then be made.

It is further recommended that the computerized analysis be extended to incorporate programs which would establish the air flow paths within variable configured sumps thus establishing where baffling or changes could be introduced to minimize the generation of flammable mixtures.

C-2

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APPENDIX

ACHE (S/T)

AUTOMATED COMBUSTION HAZARD EVALUATION

(STEADY/TRANSIENT)

Computerized analytic tools have been developed which can be used to begin correlation on experimental data for sump fires. The basic problem investigated undertakes the simulation of one dimensional flow in a duct or channel with a prescribed wall temperature distribution. The flowing fluid consists of a mixture of liquid drops in a gas phase. The gas phase consists of air as well as the vapor resulting from the evaporation of drops. The air vapor mixture, depending on temperature and concentration, may be capable of ignition and sustained combustion. Thus the problem is a complex one involving droplet size distribution, evaporation, gas phase diffusion and combustion, as well as wall and fluid heat transfer. This investigation is performed in two stages: (a) steady flow spray evaporation; and (b) transient ignition and combustion. The equations used to develop the computer codes for both, the steady flow spray evaporation model and the transient ignition and combustion model are described in the material which follows.

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STEADY FLOW SPRAY EVAPORATION MODEL

This first portion of the analysis is used to compute the droplet distribution, vapor and air concentrations, gas phase and droplet temperatures, gas phase and droplet velocities and densities as functions of axial distance for steady one dimensional flow. For this portion of the work axial conduction and diffusion are assumed to be negligible in comparison to convection and wall heat transfer effects. The equations which must be solved are those which describe the following:

1. species and phase continuity
2. droplet evaporation
3. spray equations relating droplet distribution to gas phase conditions
4. energy equation
5. droplet momentum

1. Species and Phase Continuity

The equations expressing continuity of the species as well as continuity of the phases can be written as follows:

$$\text{air} \quad m_f Y_a = m_a$$

$$\text{fuel} \quad m_f Y_{fu} + m_s = m_{fu}$$

$$\text{phases} \quad m_s + m_f = m_a + m_{fu} = m$$

where the symbols are define as:

m_f	gas phase mass flux
m_a	air mass flux (constant)
m_s	spray mass flux
m_{fu}	fuel mass flux (constant)
m	total mass flux (constant)
Y_a	air mass fraction (lb_m air/ lb_m gas phase)
Y_{fu}	fuel mass fraction (lb_m fuel/ lb_m gas phase)

All of the mass fluxes can be normalized to unity by defining the following flux fractions:

$$\begin{aligned}\psi_f &= m_f/m \\ \psi_a &= m_a/m \quad \text{constant} \\ \psi_{fu} &= m_{fu}/m \quad \text{constant} \\ \psi_s &= m_s/m\end{aligned}$$

The normalized continuity equations are:

$$\begin{aligned}\psi_f Y_a &= \psi_a \\ \psi_f Y_{fu} + \psi_s &= \psi_{fu} \\ \psi_f + \psi_s &= 1\end{aligned}$$

2. Droplet Evaporation

The droplet evaporation model used assumes the drops to be small and far enough apart so that a quasi-steady state exists between each droplet and the surrounding gas phase. The species and energy equations are solved for the local concentration and temperature field around the drop. These, together with phase equilibrium data for the liquid, are applied at the drop surface to yield a non-linear relationship between droplet temperature and gas phase temperature and concentration. As is shown in this appendix, solution of the non-linear relation at the droplet surface yields the total time rate of change of droplet radius in the following form:

$$dr/dt = -x/r = v \, dr/dx$$

where:

r droplet radius

x depends only on gas phase conditions

v droplet velocity

For the case of steady one dimensional flow the droplet radius can be obtained as a function of axial distance as:

$$r^2 + r_0^2 - 2 \int_0^x x(x')/v(x') \, dx'$$

where:

$r_0 \equiv$ inlet value of droplet radius

3. Spray Equations

The spray equations are used to compute the changes in spray mass flux and size distribution as a function of axial distance. The spray droplets are treated in groups with group index j , $j = 1, M$. Each group must have a prescribed inlet value of group droplet velocity, group droplet radius and fraction of total entering drops. The spray mass flux for each group can be defined as follows:

$$m_{sj} = \frac{4}{3} \pi \rho_l r_j^3 N_j v_j$$

where:

ρ_l absolute density of liquid

N_j number density of group j drops

v_j velocity of group j drops

For the case of steady flow, the number flux for each group of drops must remain constant, i.e., $N_j v_j = \text{constant}$, $j = 1, M$, so that the relation for mass flux of the j^{th} group can be written as follows:

$$m_{sj}/m_{sj0} = (r_j/r_{j0})^3$$

The radius ratio can be obtained from the droplet evaporation variable, x , as follows:

$$r_j/r_{j0} = (1 - \phi_j(x))^{\frac{1}{2}}$$

where:

$$\phi_j(x) = 2/r_{j0}^2 \int_0^x x(x')/v(x') dx'$$

The total spray mass flux is then given by the following:

$$m_s = \sum_{j=1, M} m_{sj}$$

The spray flux fraction is given by the following:

$$\psi_s = \sum_{j=1, M} m_{sj} / m$$

4. Energy Equation

For the flow situations under consideration, the pressure work, kinetic energy, axial diffusion and axial conduction terms can be neglected as compared with the convective and wall heat transfer terms in the energy equation. The steady one dimensional energy equation can then be written as follows:

$$d/dx (m_f h_f + m_s h_l) = K(T_w - T)$$

where:

K product of convective heat transfer coefficient and flow perimeter divided by flow area

h_f gas phase specific enthalpy

h_l liquid phase specific enthalpy

T_w wall temperature distribution

and the specific enthalpies are related to the liquid phase temperature, gas phase temperature and gas phase mass fractions as follows:

$$h_f = Y_a (h_a^0 + c_{pa} (T - T^0)) + Y_{fu} (h_{fu}^0 + c_{pfu} (T - T^0))$$

$$h_1 = h_1^0 + c_{p1} (T_1 - T^0)$$

and

h^0 enthalpy of formation at reference temperature

T^0 reference temperature 77°F

The energy equation can be rewritten in terms of non-dimensional temperature and flux fraction as follows:

$$\begin{aligned} d\theta/dx + E_1 d\psi_s/dx + E_2 \theta_{ei} d\psi_s/dx \\ - E_3 d(\psi_s \theta)/dx = E_4 (\theta_w - \theta) \end{aligned}$$

where:

$$\theta = (T - T^0)/(T_{air\ in} - T^0)$$

$$\theta_{ei} = \frac{T_{ei} - T^0}{T_{air} - T^0}$$

T_{ei} = inlet liquid temperature

T_{air} = inlet air temperature

$$\psi_c = (\psi_a c_{pa} + \psi_{fu} c_{pfu}) (T_{air\ in} - T^0)$$

$$E_1 = (h_1^0 - h_{fu}^0)/\psi_c$$

$$E_2 = (c_{p1} (T_{air\ in} - T^0))/\psi_c$$

$$E_3 = (c_{pfu} (T_{air\ in} - T^0))/\psi_c$$

$$E_4 = (K(T_{air\ in} - T^0))/(m \psi_c)$$

5. Droplet Momentum

For the relatively low velocities involved, the main effect of momentum considerations will be on the droplet group velocities, v_j , which affect the group evaporation rates. A momentum balance on a droplet of the j^{th} group yields:

$$v_j dv_j/dx = F_j$$

where the force can be deduced from the Stoke's drag law for small particles as:

$$F_j = (9/2) \mu_g / \rho_l (1/r_j^2) (u - v_j)$$

where:

μ_g gas phase viscosity
 ρ_l liquid absolute density
 r_j radius of group j droplets
 u gas phase velocity
 v_j velocity of group j droplets

In order to solve for the droplet group velocities, the gas phase velocity must also be obtained. Rewriting the phase continuity relation in terms of densities and velocities there obtains:

$$m_f = \rho_f u$$

$$m_{sj} = \rho_{sj} v_j$$

$$m_s = \sum_{j=1, M} m_{sj}$$

and

$$\rho_f u + m_s = m$$

where

ρ_f gas phase density ($\text{lb}_m \text{ gas} / \text{total ft}^3$)
 ρ_{sj} j^{th} group spray density
($\text{lb}_m \text{ group } j \text{ drops} / \text{total ft}^3$)

The gas phase density can be obtained from the absolute gas density through the following relation:

$$\rho_f = \rho_g \left(1 - \sum_{j=1, M} \rho_{sj} / \rho_l \right)$$

The absolute gas density is obtained from the perfect gas equation of state as follows:

$$\rho_g = \rho / (R T_{\text{abs}} (Y_a / W_a + Y_{fu} / W_{fu}))$$

where

ρ_g absolute gas density ($\text{lb}_m \text{ gas} / \text{ft}^3 \text{ gas}$)
 R universal gas constant
 T_{abs} absolute temperature ($^{\circ}\text{R}$)

Numerical Solution - Steady Flow Spray Evaporation Model

The computer code developed to obtain numerical solutions of the steady flow spray evaporation model is discussed below. Details of the initialization as well as the description of the wall temperature distribution are given. The finite difference equations as well as the over-all solution scheme are discussed.

Initialization

All of the input variables are defined and units are described in the comments at the beginning of the program. The input specification of the total air and spray inlet flow has been chosen to be in terms of volume flow rates (cfm). In addition, the required input for each of the droplet groups is initial radius, r_{j0} , initial velocity, v_{j0} , and fraction of total drops, f_{pj} . Amplification of the initialization of the mass fluxes, densities, gas phase velocity and number densities are given below.

The inlet mass fluxes of air and total spray can be obtained from the input values of volume flow rate and flow area as:

$$\begin{aligned}\dot{m}_a &= 0.075 \dot{V}_a / 60 / A \\ \dot{m}_s &= \rho_l \dot{V}_s / 60 / A\end{aligned}$$

where

$$\begin{aligned}\dot{V}_a & \text{ inlet volume flow rate of air (cfm)} \\ \dot{V}_s & \text{ inlet volume flow rate of spray (cfm)} \\ A & \text{ flow area of channel (ft}^2\text{)}\end{aligned}$$

The total mass flux of spray can also be written in terms of the inlet total droplet number density as:

$$m_s = \sum_{j=1,M} \frac{4}{3} \pi r_{j0}^3 \rho_l f_{pj} N_t v_{j0}$$

The total number density of drops at the inlet can then be obtained from:

$$N_t = m_s / \left(\frac{4}{3} \pi \rho_l \sum_{j=1,M} r_{j0}^3 f_{pj} v_{j0} \right)$$

The individual group number densities can then be obtained as

$$N_{j0} = N_t f_{pj}$$

and the individual group mass fluxes are given by:

$$m_{sj0} = \frac{4}{3} \pi \rho_l r_{j0}^3 N_{j0} v_{j0}$$

The inlet value of the gas phase mass flux is assumed to be equal to the inlet value of the air mass flux, i.e.,

$$m_{f0} = m_a$$

The total inlet mass flux is obtained by adding the inlet total spray and gas phase mass fluxes and the inlet values of the flux fractions are obtained by simple division. The inlet value of the gas phase velocity is obtained by applying the equations previously shown for the droplet momentum calculations.

Wall Temperature Distribution

The wall temperature distribution is input by specifying the value of wall temperature, $TWALL(I)$, at consecutive axial locations, $XWALL(I)$, for all of the wall points, $I = 1, NWALL$. $XWALL(1)$ is the axial position of the inlet and $XWALL(NWALL)$ is the axial position of the end of the channel.

Numerical Scheme and Equations

The axial length of the channel is broken up into increments of length Dx . The continuity relations, an integrated form of the energy equation, droplet evaporation equations and the spray equations are applied at each new station down the length of the channel. After satisfying these equations simultaneously at a new station, the droplet momentum and phase continuity relations are used to obtain the new droplet group velocities and gas phase velocity. Simultaneous solution of the continuity, energy, evaporation, and spray equations at each new station requires an iterative solution using the secant method. This method is also applied to solve for the droplet group and phase velocities. The nonlinear relations between spray flux fraction, ψ_s , gas phase temperature, θ , and liquid drop temperature, θ_l , which must be satisfied at each new station are given below. Using the superscript $(^n)$ to denote the values of a dependent variable at a new station, the equations are as follows:

Continuity:

$$\psi_f^n + \psi_s^n = 1$$

$$\psi_f^n Y_a^n + \psi_a = \text{constant}$$

$$\psi_f^n Y_{fu}^n + \psi_s^n = \psi_{fu} = \text{constant}$$

Energy:

$$\begin{aligned} \theta^n - \theta + E_1 (\psi_s^n - \psi_s) + E_2 \theta_{ei} (\psi_s^n - \psi_s) \\ - E_3 (\psi_s^n \theta^n - \psi_s \theta) = E_4 Dx/2 (\theta_w^n + \theta_w - \theta^n - \theta) \end{aligned}$$

Evaporation:

$$x^n = x^n (\theta^n, \theta_1^n, Y_{fu}^n)$$

Spray:

$$\psi_s^n = \sum_{j=1, M} m_{sj}^n / m$$

$$m_{sj}^n = m_{sj0} rr_j^3$$

$$rr_j = (1 - \theta_j^n)^{1/2}$$

$$\phi_j^n = \phi_j + (x^n + x) Dx / r_{j0}^2 / v_j$$

where

rr_j = ratio of radius to inlet radius for group j.

The use of the secant method requires initial guesses for the spray flux fraction, ψ_s^n , at the new station. With the initial guess, subroutine MFLUX is called to evaluate the other flux

fractions and mass fractions. Subroutine ECOEF is called to obtain the coefficients of the non-dimensional gas phase and liquid temperatures which are then used by subroutine EVAP to calculate these temperatures as well as the evaporation rate parameter, x^n . The evaporation rate parameter is then used in subroutine SPRAY to calculate the spray flux fraction. The calculated spray flux fraction is compared to the initial guess and the secant method is used to obtain a new guess. This process is continued up to 50 times (in which case a non-convergence message is printed) unless convergence to within .0005 is attained for the spray flux fraction. Once the secant method has converged for the flux fractions, mass fractions and temperatures at the new station, the droplet and gas phase velocities are calculated.

The droplet group velocities and gas phase velocity are calculated at the new station using the fact that the gas phase flux fraction and droplet group mass fluxes, m_{sj}^n , have already been determined at the new station. The droplet momentum equation (section 5) is written in finite difference form using a forward difference for the velocity derivative and values of velocity at the new station for the drag term on the right hand side. There results the following equation:

$$v_j^n = (B_j u^n + v_j^2) / (v_j + B_j)$$

which relates the group velocities to the gas phase velocities at the new station. The group dependent parameter is defined as

$$B_j = (\frac{9}{2} \mu_g D_x) / (\rho_1 r_j^2)$$

The droplet group densities can be calculated in terms of the known mass fluxes and new group velocities as

$$\rho_{sj}^n = m_{sj}^n / v_j^n$$

The absolute gas density at the new station, ρ_g^n , is calculated from the perfect gas relation as

$$\rho_g^n = p / (R T_{abs}^n (Y_a^n / W_a + Y_{fu}^n / W_{fu}))$$

The gas phase density is obtained from the absolute gas phase density and spray droplet group densities at the new station as:

$$\rho_f^n = \rho_g^n (1 - \sum_{j=1, M} \rho_{sj}^n / \rho_1)$$

The gas phase velocity at the new station can be obtained from the known gas phase flux fraction as

$$u^n = m \psi_f^n / \rho_f^n$$

Since the system of equations which must be solved to obtain new values of gas phase velocity and group velocities are non-linear, the secant method is again applied to solve first for the gas phase velocity at the new station. Initial guesses are made for the gas phase velocity at the new station and the system of equations is used to generate u^n . Subsequent guesses are generated by the secant method until convergence results.

This completes one cycle in the calculations. The values generated for the dependent variables at the new station are then placed in storage locations for the old values and a new cycle is performed. This continues until either the end of the channel is reached or all of the droplets have evaporated. If all drops have evaporated at any particular station in the flow, the only equation solved throughout the remainder of the channel is the energy equation with $\psi_s = 0$. The change in gas density and velocity are also calculated.

TRANSIENT IGNITION AND COMBUSTION MODEL

For the second phase of the investigation, a transient ignition and combustion model together with numerical code have been developed. The purpose of the transient ignition and combustion model is to determine whether the conditions at any point in the flow channel as determined from the steady flow spray evaporation code are such as to allow ignition and sustained combustion to occur. The model developed takes into account axial diffusion and conduction as well as kinetically controlled chemical reaction, which is assumed to occur in the gas phase, and droplet size distribution and evaporation. The model uses a Lagrangian co-ordinate system fixed to the flow wherein convective terms are neglected. In this co-ordinate system, the energy supplied by an ignition source over a given time duration, is assumed to result in an elevated gas phase temperature which appears as a step in the initial temperature distribution. The length of the step is equal to the product of the gas phase velocity at the axial location of the ignition source and the time duration of the source. The gas phase velocity as well as all other initial conditions for the transient model are obtained from the steady spray evaporation code at the axial position considered. The height of the temperature step is determined by the amount of energy deposited by the ignition source into the gas phase. With this temperature step as an initial condition, the model employs an Arrhenius form of

reaction rate together with transient one-dimensional species equations for fuel and air and the transient one-dimensional energy equation as well as droplet evaporation and spray equations to obtain the spatial distributions of spray, fuel, air and product mass fractions and gas phase temperature as functions of time. The resulting distributions can then be examined to determine whether ignition can occur and whether flame propagation upstream and sustained combustion can occur. The equations used in the transient ignition and combustion model are shown below.

The one-dimensional transient species equations for the fuel and air are:

$$Y_f \frac{\partial Y_{Fu}}{\partial t} = \frac{1}{\rho_i} \frac{\partial}{\partial x} (\rho_f D \frac{\partial Y_{Fu}}{\partial x}) + \frac{\omega_{Fu}}{\rho_i} + (1 - Y_{Fu}) \frac{\partial Y_f}{\partial t}$$

$$Y_f \frac{\partial Y_A}{\partial t} = \frac{1}{\rho_i} \frac{\partial}{\partial x} (\rho_f D \frac{\partial Y_A}{\partial x}) + \frac{\omega_A}{\rho_i} - Y_A \frac{\partial Y_f}{\partial t}$$

where

- Y_f gas phase mass fraction (lb_m gas/total lb_m)
- Y_{Fu} fuel mass fraction (lb_m gaseous fuel/lb_m gas)
- Y_A air mass fraction (lb_m air/lb_m gas)
- ρ_i initial total density as obtained from steady flow spray evaporation code
- $\rho_f D$ product of gas phase density and mass diffusivity
- ω_{Fu} rate of generation of fuel from chemical reaction
- ω_A rate of generation of air from chemical reaction

For the model, the Lewis number is assumed to be unity so that

$$\rho_f D = \frac{\lambda}{C_{pref}}$$

where

λ mixture thermal conductivity (air)

C_{pref} mixture specific heat (average of gas phase components)

The reaction rates are assumed to be of the Arrhenius form as

$$\omega_{Fu} = - Y_{Fu} Y_A k e^{-E/RT}$$

$$\omega_A = \text{STC} \quad Fu$$

where

k is the pre-exponential factor

E is the activation energy

R is the universal gas constant

STC is the stoichiometric coefficient corresponding to ($1b_m$ air burned/ $1b_m$ fuel)

The species equations for fuel and air can be combined into a single equation for a so-called mixture fraction defined as

$$\Sigma = Y_A / \text{STC} - Y_{Fu}$$

The differential equation for the mixture fraction does not involve a reaction rate term and is

$$Y_f \frac{\partial \Sigma}{\partial t} = \frac{1}{\rho_i} \frac{\partial}{\partial x} \left(\rho_f D \frac{\partial \Sigma}{\partial x} \right) - \xi \frac{\partial Y_f}{\partial t} - \frac{\partial Y_f}{\partial t}$$

The value of the mixture fraction is calculated at each position for each time step and is used to determine which of the species limits the reaction. If the mixture fraction is negative, then the air mass fraction is limiting and the air species equation is solved to determine Y_A which together with the already calculated mixture fraction yields the fuel mass fraction. If the mixture fraction is positive, the fuel species equation is used. The product mass fraction is then simply obtained since the sum of fuel, air and product mass fractions must add to unity.

The one-dimensional transient energy equation neglecting convective terms in the Lagrangian co-ordinates and also neglecting kinetic energy and pressure work terms is as follows:

$$\frac{\partial h}{\partial t} = \frac{\lambda}{\rho_i} \frac{\partial^2 T}{\partial x^2} + \frac{1}{\rho_i} \frac{\partial}{\partial x} \sum_k h_k \rho_f D \frac{\partial Y_k}{\partial x} + \frac{K}{\rho_i} (T_\omega - T)$$

where

K = product of convective heat transfer coefficient and flow perimeter divided by flow area

the subscript k corresponds to

$k = 1$ for gaseous fuel

$k = 2$ for air

$k = 3$ for products

and

$$h = (1 - Y_f) h_s + Y_f h_f$$

$$h_s = h_\ell^o + C_{p\ell} (T_\ell - T_o)$$

$$h_f = \sum_k Y_k h_k$$

$$h_k = h_k^o + C_{pk} (T - T_o)$$

$$h^o = \text{enthalpy of formation, } T_o = 77^\circ\text{F}$$

The energy equation can be rewritten in terms of non-dimensional temperature and enthalpy as:

$$\begin{aligned} \frac{\partial \bar{h}}{\partial t} = & \frac{\lambda (T_{\text{step}} - T_i)}{\rho_i h_{\text{ref}}} \frac{\partial^2 \Theta}{\partial x^2} + \frac{\lambda}{\rho_i C_{p\text{ref}}} \frac{\partial}{\partial x} \sum_k (H_k + HT_k \Theta) \frac{\partial Y_k}{\partial x} \\ & + \frac{K(T_{\text{step}} - T_i)}{i h_{\text{ref}}} (\Theta_\omega - \Theta) \end{aligned}$$

where:

$$\bar{h} = \frac{h}{h_{\text{ref}}} = Y_s (H_1 + H_2 \Theta_{ei}) + (1 - Y_s) \sum_k Y_k (H_k + HT_k \Theta)$$

$$h_{\text{ref}} = \sum_k [h_k^o + C_{pk} (T_i - T_o)]$$

$$Y_s = 1 - Y_f = \text{spray mass fraction (lb}_m \text{ spray/total lb}_m \text{)}$$

$$H_1 = [h_\ell^o + C_{p\ell} (T_i - T_o)]/h_{\text{ref}}$$

$$H_2 = C_{p\ell} (T_{\text{step}} - T_i)/h_{\text{ref}}$$

$$H_k = [h_k^o + C_{pk} (T_i - T_o)]/h_{\text{ref}}$$

$$HT_k = [C_{pk} (T_{\text{step}} - T_i)]/h_{\text{ref}}$$

$$\theta = (T - T_i)/(T_{\text{step}} - T_i)$$

$$\theta_{ei} = (T_{ei} - T_i)/(T_{\text{step}} - T_i)$$

T_i = initial temperature outside of step

T_{ei} = inlet liquid temperature

T_{step} = temperature of step due to ignition source

subscript denotes values for liquid drops

The evaporation rate parameter, χ , is calculated in exactly the same way as for the steady flow evaporation model. The spray mass fraction, however, is given by

$$Y_s(x,t) = \sum_{j=1,M} Y_{sj}(x,0) RR_j^3(x,t)$$

$$RR_j(x,t) = [1 - \phi_j(x,t)]^{1/2}$$

$$\phi_j(x,t) = 2 \int_0^t \frac{\chi(x,t') dt'}{r_{j0}^3(x,0)}$$

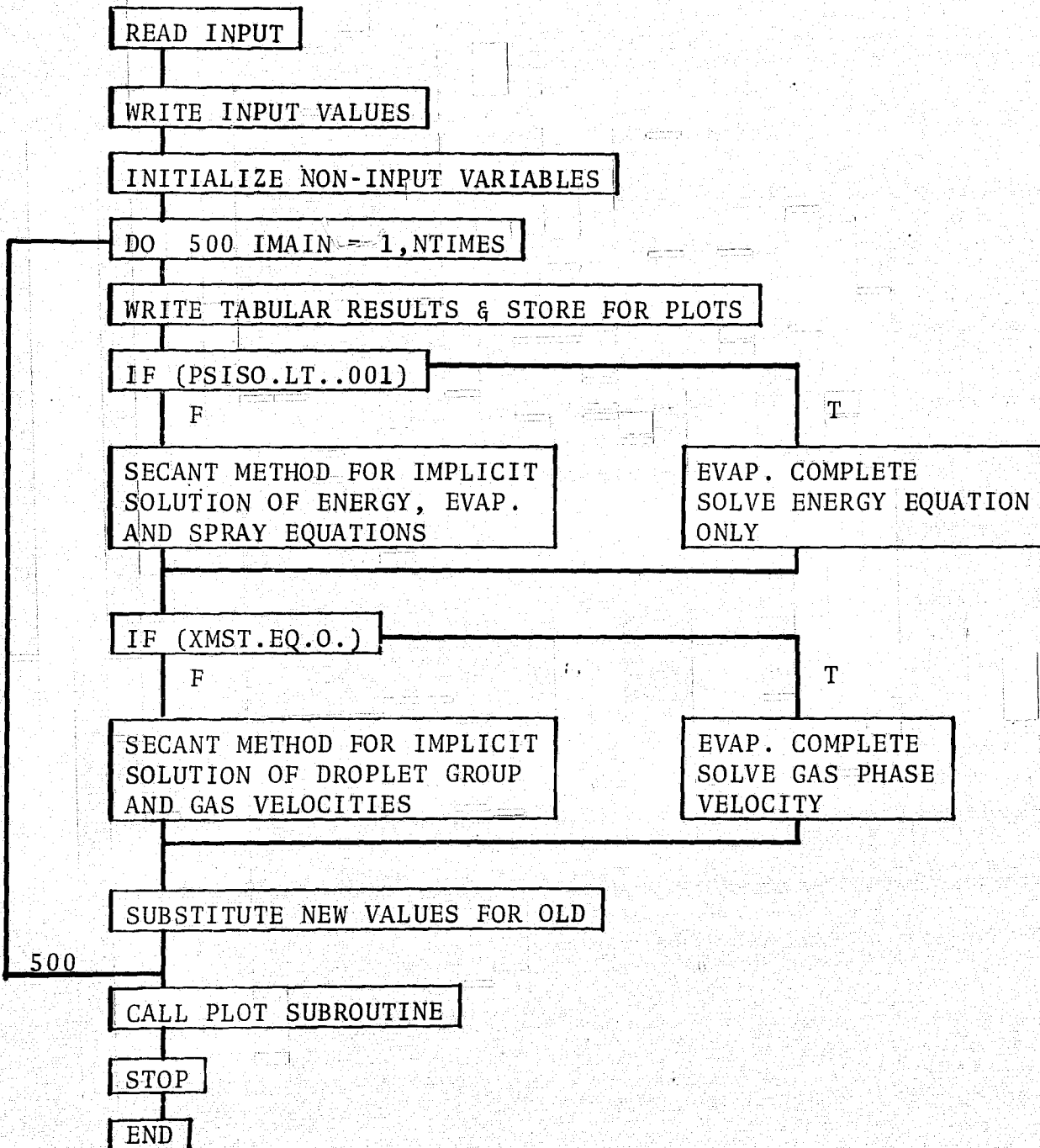
Numerical Solution of Transient Ignition and Combustion Model

In order to obtain solutions within reasonable computer time limits, the computer code has been developed using an explicit numerical scheme. The time step and uniform value of node spacing are read in as input. The code explicitly calculate the values of the dependent variables at the new time in terms of the values at the previous time step. Starting from the specified initial conditions, a new value of evaporation parameter is calculated using old values of gas phase temperature and fuel mass fraction. This new evaporation parameter is used along with the old in a trapezoidal rule integration to obtain the new parameter, ϕ_j . This new value of ϕ_j is used to calculate the new value of spray mass fraction, Y_s . This new value of spray mass fraction is used in the mixture fraction equation as well as either the fuel or air species equations to calculate new values of fuel, air, and product mass fractions. In order to insure stability of the species equations involving the possibly very large reaction rates an exponential approximation is used. This approximation remains stable for high rates of reaction and reduces to the regular explicit finite difference equations in the event of slow reaction. Once the new values of spray, fuel, air and product mass fraction have been determined for the nodal point, the energy equation is used to determine the new value of temperature. At each time step, the above occurs for every nodal point.

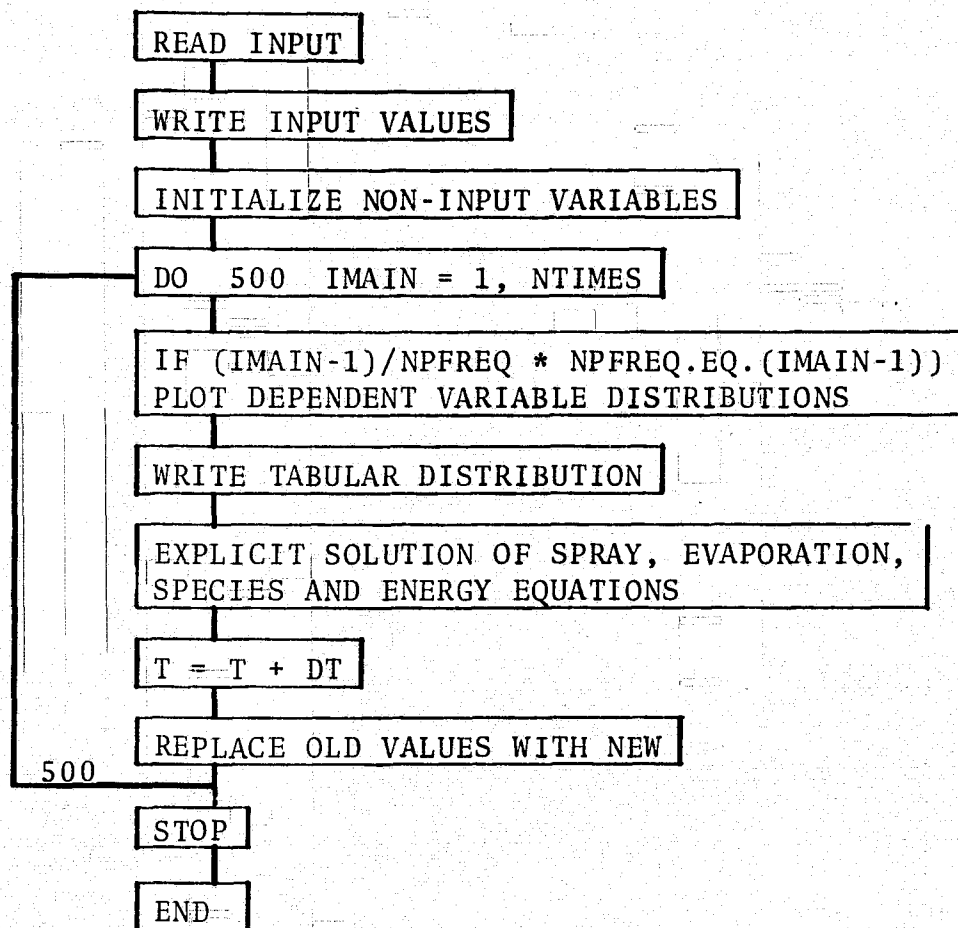
FLOW CHARTS AND GEOMETRY

The numerical flow charts for the two codes are found on the next two pages followed by a drawing showing the geometry of the transient ignition of combustion model.

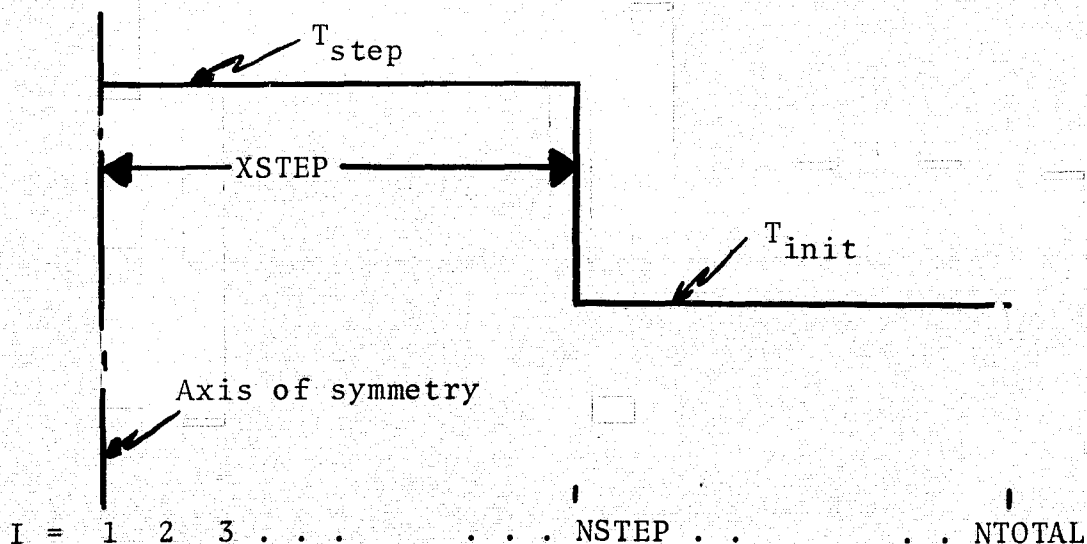
STEADY FLOW SPRAY EVAPORATION



TRANSIENT IGNITION AND COMBUSTION



GEOMETRY FOR TRANSIENT IGNITION AND COMBUSTION



SAMPLE PROBLEMS

The fuel chosen for the sample problems is n-decane for both the steady and transient codes. A discussion of the input and output for each of these cases is given below.

Steady Flow Spray Evaporation

All of the properties for air, fuel and droplets must be input. All input data must have field width of 10. All input variables as well as units are given in the comments at the beginning of each of the two codes. All input data are printed after being read.

The first card of input specifies the air properties of specific heat, inlet air temperature, molecular weight as well as mixture thermal conductivity and viscosity (taken as air values) all with field width of 10.

The second card specifies the fuel properties, specific heats of gaseous and liquid fuel, enthalpies of formation of gaseous and liquid fuel, molecular weight, inlet liquid temperature and absolute liquid density.

The third card of input contains the total number of droplet groups (MGROUP) in I10 format.

The next MGROUP cards each contains three values required for each group, initial radius, fraction of particles and inlet velocity.

The next card contains two data points of vapor pressure-temperature data as well as the system pressure.

The next card contains the volumetric flow rate of air and spray, flow area, wall heat transfer coefficient and length of flow perimeter.

The number of wall points used to characterize the wall temperature distribution is read in next as NWALL in I10 format.

The next NWALL cards contain the wall temperature distribution information. Each card must contain one wall position and the wall temperature corresponding to that position. The first value of wall position, XWALL(1), is taken to be the inlet position. The final wall position, XWALL(NWALL), must be larger than the axial distance over which the calculations are to be carried out.

The next card specifies the length of the axial position increments to be used in the calculations.

The final card specifies the total number of steps to be taken. Use a value which is one larger than the total number of increments.

The input values are echo-checked as the first page of output, where the fortran names and appropriate units are printed together with the numerical values.

The sample problem is for n-decane drops. There is only one droplet group with initial radius of 30 microns, group fraction of 1. and droplet group inlet velocity of 0.5 fps.

The inlet air flow rate is 30 scfm and the liquid inlet flow rate is 0.03 cfm. The duct cross-sectional area is 1 ft² and the perimeter is 4 ft. The inlet air and droplet temperatures are 500°F and 200°F, respectively while the wall temperature is constant at 400°F. The wall heat transfer coefficient is 20 Btu/hr/ft²/°F. The increment in the axial direction is 0.01 ft.

The printed output consists of position, air and fuel mass fraction in the gas phase, total density, gas, wall and liquid drop temperatures, gas velocity and droplet group radius and group spray mass fraction. In addition to the tabulated results, line printer plots are also provided for all of the output variables except the droplet group information.

From the plots it can be seen that for the inlet conditions of this sample problem, evaporation occurs relatively quickly, within the first 4 steps. The air mass fraction which begins at an inlet value of unity (with no fuel vapor in the gas) decreases as evaporation proceeds to a constant value of 0.621. The fuel mass fraction which has an inlet value of zero increase to a value of 0.379 within the first 4 steps. Both the air and fuel mass fractions are for the gas phase, i.e., lbm of air and gaseous fuel per lbm of gas. The total density for the constant pressure process decreases continuously. The initial rapid decrease in density is due to the evaporation of the drops counteracted somewhat by the rapid decrease in gas temperature.

The slow decrease in density after evaporation is complete is due to the rising temperature of the gas phase. The gas temperature drops rapidly due to the heat required for vaporization and once vaporization is complete, the gas temperature increases due to wall heat transfer to approach the wall temperature.

Transient Ignition and Combustion Model

As in the case of the steady code, all properties are read in as input and printed out. All input variables are defined with units in comment cards at the beginning of the program. In addition, the initial temperature distribution must be specified depending on the ignition source duration and energy and gas phase velocity.

For the sample problem a mixture of n-decane vapor and air together with drops of n-decane at a temperature of 500°F, total density of 0.08 lbm/ft³ air mass fraction in gas phase of 0.80 and gas phase n-decane (fuel) mass fraction of 0.20. The geometry is specified in the following way. Taking the product of gas velocity and ignition source duration, the total step length is calculated. For our example, let us say this product was equal to 0.2 ft, e.g., 20 ft/sec gas velocity and 0.01 sec ignition source time duration. This yields a step half width of XSTEP 0.1 ft. The initial temperature of the step is then calculated from the energy supplied by the ignition source. Assuming all of the energy (6 Btu) is used to heat only the gas phase, this results in a temperature rise of 1500°F calculated from:

$$\Delta T = \frac{Q_{in}}{2 \cdot XSTEP \cdot A \cdot \rho_f (Y_{Fu} C_{pfu} + Y_{air} C_{pair})}$$

$$\rho_f = Y_f \rho_i$$

$$Y_f = 1 - Y_s = 1 - \sum Y_{sj}$$

So that $T_{init} = 500^\circ\text{F}$ and $T_{step} = 2000^\circ\text{F}$. The wall temperature is assumed to be 400°F . The time step size, total number of time steps and plot frequency are all read in as program control variables. For the sample problem, the time step was chosen to be 0.01 sec, 50 time steps were executed and the plot frequency was 10.

The output consists of tabulated distributions of spray fraction, i.e., lbm spray/total lbm, non-dimensional temperature, fuel mass fraction, air mass fraction and product mass fraction at each time step. In addition to the tabulated results at each time step, every 10 steps line printer plots are produced. From these results, it is seen that for this sample problem the spray evaporates in the first time step. Also, within the high temperature step, complete combustion occurs such that all of the air in this region is burned and the temperature increases, while outside the temperature step, the gas temperature is initially too low to cause any appreciable reaction to occur and the temperature decreases in order to supply energy to evaporate the drops. It can be noted here that if the ignition source energy is very low, then the reaction rate will be very low and the

initial temperature step may decay due to conduction and wall heat transfer, i.e., ignition may not occur. Also, it may be seen that the initial mixture is rich and the air mass fraction limits the energy released by combustion. After the initial ignition phase, the field can be clearly seen to consist of a "burned" zone composed of fuel and products and an "unburned" zone composed of air and fuel. These two zones are separated by a "flame zone" in which combustion, conduction and diffusion occur. Once ignition has occurred, it is the motion of this "flame zone" through the gas which determines whether sustained combustion can occur. If the speed of propagation calculated from the results of the transient ignition and combustion model is larger than the gas phase velocity at the location of interest (obtained from the steady evaporation code) then sustained combustion is predicted. On the other hand, if the "flame zone" propagation speed is less than the gas phase velocity, then ignition but unsustained combustion is predicted. The latter case is predicted in this sample case.

DROPLET EVAPORATION (SPHERICALLY SYMMETRIC)

A quasi-steady approximation is used to determine the droplet evaporation rate.

Species Equation For Fuel (in gas)

$$\frac{d}{dr} (r^2 \rho_f v Y_{Fu}) = \frac{d}{dr} [r^2 \rho_f D \frac{dY_{Fu}}{dr}]$$

Energy Equation (in gas)

$$\frac{d}{dr} (r^2 \rho_f v C_p (T-T_o)) = \frac{d}{dr} [r^2 \frac{\lambda}{C_p} \frac{\partial}{\partial r} C_p (T-T_o)]$$

The mass flow rate leaving drop, $\dot{m}$, is

$$\dot{m} = \rho_f v_\ell 4 \pi r_\ell^2, \quad v = \text{radial velocity}$$

Let

$$\xi(r) = \frac{\dot{m}}{\rho_f D} \int_r^\infty \frac{1}{4 \pi r^2} dr$$

$$d\xi(r) = - \frac{\dot{m}}{\rho_f D 4 \pi r^2} dr$$

For

$$Le = \frac{\lambda}{\rho_f C_p D} = 1, \quad C_p = \text{constant}$$

$$\frac{dY_{Fu}}{d\xi} = - \frac{d^2 Y_{Fu}}{d\xi^2}$$

$$\frac{dT}{d\xi} = - \frac{d^2 T}{d\xi^2}$$

Solving

$$Y_{Fu} = A_{Fu} + B_{Fu} e^{-\xi}$$

$$T = A_T + B_T e^{-\xi}$$

Energy Balance at Surface of Drop

$$\frac{4\pi r^2 \lambda}{\dot{m}} \left. \frac{dT}{dr} \right|_{r=r_\ell} = L'$$

where

$$L' = h_{Fu} - h_\ell + C_{p\ell} (T_\ell - T_{\ell i})$$

$$h_{Fu} = h_{Fu}^o + C_{pFu} (T_\ell - T_o)$$

$$h_\ell = h_\ell^o + C_{p\ell} (T_\ell - T^o)$$

$T_{\ell i}$ = inlet liquid temperature

T_ℓ = liquid drop interface temperature

or

$$-\left. \frac{dT}{d\xi} \right|_{\xi=\xi_\ell} = \frac{L}{C_p} \quad \text{for } Le = 1$$

Apply B.C.'s

$$\text{as } r \rightarrow \infty \quad (\xi \rightarrow 0) \quad \left. \begin{array}{l} Y_{Fu} \rightarrow Y_{Fu\infty} \\ T \rightarrow T_\infty \end{array} \right\} \text{Gas Phase Values}$$

$$A_{Fu} + B_{Fu} = Y_{Fu\infty}$$

$$A_T + B_T = T_\infty$$

$$\left. \begin{array}{l} \text{at } r = r_\ell \ (\xi = \xi_\ell) \\ T = T_\ell \\ Y_{Fu} = Y_{Fu\ell} \end{array} \right\} \text{from vapor pressure data}$$

also at $r = r_\ell \ (\xi = \xi_\ell)$, No net flow of components other than fuel $(1 - Y_{Fu})$

i.e.

$$[\rho v (1 - Y_{Fu}) - \rho D \frac{d(1 - Y_{Fu})}{dr}]_{r=r_\ell} = 0$$

or

$$[1 - Y_{Fu} + \frac{d(1 - Y_{Fu})}{d\xi}]_{\xi_\ell} = 0$$

$$[Y_{Fu} + \frac{dY_{Fu}}{d\xi}]_{\xi_\ell} = 1$$

Then

$$A_{Fu} + B_{Fu} e^{-\xi_\ell} = Y_{Fu\ell}$$

$$A_T + B_T e^{-\xi_\ell} = T_\ell$$

$$A_{Fu} + B_{Fu} e^{-\xi_\ell} - B_{Fu} e^{-\xi_\ell} = 1 \quad (A_{Fu} = 1)$$

Solve for A_T and B_T

$$A_T = T_\infty - B_T$$

$$T_\infty - B_T (1 - e^{-\xi_\ell}) = T_\ell$$

$$B_T = \frac{T_\infty - T_\ell}{1 - e^{-\xi_\ell}} \quad A_T = T_\infty - B_T$$

$$\frac{dT}{d\xi} = -B_T e^{-\xi} = -\left(\frac{T_\infty - T_\ell}{1 - e^{-\xi_\ell}}\right) e^{-\xi}$$

Solve for A_{Fu} , B_{Fu}

$$A_{Fu} = Y_{Fu_{\infty}} - B_{Fu}$$

$$Y_{Fu_{\infty}} - B_{Fu} + B_{Fu} e^{-\xi_l} = Y_{Fu_l}$$

$$B_{Fu} = \frac{Y_{Fu_{\infty}} - Y_{Fu_l}}{1 - e^{-\xi_l}}$$

$$A_{Fu} = Y_{Fu_{\infty}} - \frac{Y_{Fu_{\infty}} - Y_{Fu_l}}{1 - e^{-\xi_l}} = 1$$

or

$$Y_{Fu_{\infty}} - Y_{Fu_{\infty}} e^{-\xi_l} - Y_{Fu_{\infty}} + Y_{Fu_l} = (1 - e^{-\xi_l})$$

$$(1 - Y_{Fu_{\infty}}) e^{-\xi_l} = 1 - Y_{Fu_l}$$

$$e^{-\xi_l} = \frac{1 - Y_{Fu_l}}{1 - Y_{Fu_{\infty}}}$$

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Energy Balance at Interface Yields

$$\frac{T_{\infty} - T_l}{1 - e^{-\xi_l}} e^{-\xi_l} = \frac{L}{C_p}$$

These two equations together with vapor pressure data must be solved to obtain the evaporation rate.

Evaporation Rate (R)

$$R \equiv \frac{dr_l}{dt} \equiv - \frac{x}{r_l}$$

$$\dot{m} = -4\pi r_l^2 \rho_l \frac{dr_l}{dt} \quad \text{and} \quad \xi_l = \frac{\dot{m}}{4\pi \rho_f D} \frac{1}{r_l}$$

$$\frac{dr_l}{dt} = \frac{\dot{m}}{4\pi r_l^2 \rho_l} = -\xi_l \frac{\rho_f D}{\rho_l r_l}$$

$$\text{For } Le = 1 \quad \rho_f D = \frac{\lambda}{C_p}$$

$$R = \frac{dr_l}{dt} = \frac{\lambda}{\rho_l C_p} \frac{\xi_l}{r_l} = \frac{-x}{r_l}$$

$$x = + \frac{\lambda}{\rho_l C_p} \xi_l$$

But

$$\xi_l = \ln \left[\frac{1 - Y_{Fu_\infty}}{1 - Y_{Fu_l}} \right]$$

$$x = \frac{\lambda}{\rho_l C_p} \cdot \ln \left[\frac{1 - Y_{Fu_\infty}}{1 - Y_{Fu_l}} \right]$$

Vapor Pressure Data

From flammability diagram for Jet II-lubricant, the partial pressure of lubricant (fuel) vapor can be approximated by

$$\ln P_{Fu} = AT + B$$

From two data points $(P_{F1}, T_{l2}), (P_{F2}, T_{l2})$

$$A = \frac{1}{T_{l1} - T_{l2}} \ln \left(\frac{P_{F1}}{P_{F2}} \right)$$

$$B = \ln (P_{F1}) - AT_{l1}$$

The air to fuel ratio (STC) is assumed large so that the molecular weight of products formed is close to that of air and the fuel mass fraction can be related to partial pressure as

$$Y_{Fu_l} = \frac{P_{Fu_l} W_{Fu}}{P_{Fu_l} W_{Fu} + P_A W_A + P_{pr} W_{pr}}$$

If $W_{pr} \approx W_{air}$

$$Y_{Fu_l} = \frac{P_{Fu_l} W_{Fu}}{P_{Fu_l} W_{Fu} + (P_A + P_{pr}) W_{air}}$$

Since $P_{Fu} + P_A + P_{pr} = P_T$ total pressure

$$Y_{Fu\ell} = \frac{P_{Fu\ell} W_{Fu}}{P_{Fu\ell} W_{Fu} + (P_T - P_{Fu\ell}) W_{air}}$$

This equation together with the vapor pressure vs. liquid temperature equation yields the dependence of fuel mass fraction at droplet surface to surface temperature, i.e.,

$$P_{Fu\ell} = e^{AT_\ell + B}$$

and combined, these yield, $Y_{Fu\ell} = Y_{Fu\ell}(T_\ell)$ which can be used to satisfy the interface conditions derived earlier, i.e.,

$$e^{-\xi_\ell} = \frac{1 - Y_{Fu\ell}}{1 - Y_{Fu\infty}}$$

$$\frac{T_\infty - T_\ell}{1 - e^{-\xi_\ell}} e^{-\xi_\ell} = \frac{L}{C_p}$$

Once a solution is obtained (iterative) the T_ℓ , $Y_{Fu\ell}$, ξ_ℓ are known for the given gas phase conditions T_∞ , $Y_{Fu\ell}$ and the drop evaporation rate is given by

$$R = \frac{dr_\ell}{dt} = - \frac{X}{r} \quad X = \frac{\lambda}{\rho_\ell C_p} \ln \left[\frac{1 - Y_{Fu\infty}}{1 - Y_{Fu\ell}} \right]$$

Fuel and Air Species Equations for Transient Model

$$Y_f \frac{\partial Y_{Fu}}{\partial t} = \frac{1}{\rho_i} \frac{\partial}{\partial x} (\rho_f D \frac{\partial Y_{Fu}}{\partial x}) + \frac{\omega_{Fu}}{\rho_i} + (1 - Y_{Fu}) \frac{\partial Y_f}{\partial t}$$

$$Y_f \frac{\partial Y_A}{\partial t} = \frac{1}{\rho_i} \frac{\partial}{\partial x} (\rho_f D \frac{\partial Y_A}{\partial x}) + \frac{\omega_A}{\rho_i} - Y_A \frac{\partial Y_f}{\partial t}$$

where $\omega_{Fu} = - Y_{Fu} Y_A k e^{\frac{-E}{RT}} = \text{Arrhenius reaction rate}$

$\omega_A = \text{STC} \cdot \omega_{Fu}$

Define mixture fraction $\xi = \frac{Y_A}{\text{STC}} : Y_{Fu}$

Two equations can be combined to yield:

$$Y_f \frac{\partial \xi}{\partial t} = \frac{1}{\rho_i} \frac{\partial}{\partial x} (\rho_f D \frac{\partial \xi}{\partial x}) - \xi \frac{\partial Y_f}{\partial t} - \frac{\partial Y_f}{\partial t}$$

or

$$\frac{\partial}{\partial t} (Y_f \xi) = \frac{1}{\rho_i} \frac{\partial}{\partial x} (\rho_f D \frac{\partial \xi}{\partial x}) - \frac{\partial Y_f}{\partial t}$$

Then using a forward difference in time and central difference for the diffusion term:

$$Y_{fN} \xi_N - Y_{fO} \xi_O = \frac{(\rho_f D) \Delta t}{\rho_i (\Delta x)^2} [\xi_O(i+1) + \xi_O(i+1) - 2\xi_O(i)] - (Y_{fN} - Y_{fO})$$

$$\xi_N = \frac{1}{Y_{fN}} (Y_{fO} \xi_O + E_2 \cdot DT \cdot [\xi_O(i+1) + f_O(i+1) - 2f_O(i)] + Y_{fO} - Y_{fN})$$

$$\xi = \frac{Y_A}{STC} - Y_{Fu}$$

Complete solution requires solution for ξ which does not involve reaction rate and either solution for Y_A or Y_{Fu} . For high temperatures, the rate of reaction is limited by the decreasing concentration of air and fuel. If $\xi_N < 0$, then the air mass fraction will limit the reaction so that Y_A will be solved for. If $\xi_N > 0$, then fuel mass fraction limits the reaction and Y_{Fu} is solved for.

For $\xi_N > 0$

$$Y_f \frac{\partial Y_{Fu}}{\partial t} = E_2 (Y_{FuO}(i+1) + Y_{FuO}(i+1) - 2Y_{FuO}(i+1)) + \frac{\partial Y_f}{\partial t} - Y_{Fu} \frac{\partial Y_f}{\partial t} - \frac{Y_{Fu} Y_A k_e}{\rho_i} e^{-\frac{E}{RT_O}}$$

$$\frac{\partial Y_{Fu}}{\partial t} = \frac{E_2}{Y_f} \phi Y_{FuO} + \left(\frac{1}{Y_f} \frac{\partial Y_f}{\partial t} \right) - Y_{Fu} \left(\left(\frac{1}{Y_f} \frac{\partial Y_f}{\partial t} \right) + \frac{Y_{Ao} k_e}{\rho_i Y_f} \right) e^{-\frac{E}{RT_O}}$$

where $Y_f = \frac{Y_{fN} + Y_{fO}}{2}$

$$\left(\frac{1}{Y_f} \frac{\partial Y_f}{\partial t} \right) = \frac{1}{DT} \ln \left(\frac{Y_{fN}}{Y_{fO}} \right)$$

Exponential Approximation

In order to treat the high reaction rates at high temperatures, an exponential approximation is applied to the limiting mass fraction (Y_{Fu} for $\xi > 0$; Y_A for $\xi < 0$) so that the rapid decrease in mass fraction and subsequent decrease in rate can be adequately modeled.

Assume $Y_{Fu} = A_{Fu} + B_{Fu} e^{-C_{Fu} t}$ applied during time step
 at T $Y_{Fu} = Y_{Fu0}$ $A_{Fu} + B_{Fu} = Y_{Fu0}$

For $\xi_N > 0$

$$Y_{FuN} = Y_{Fu0} e^{-C_{Fu} t} + A_{Fu} (1 - e^{-C_{Fu} t})$$

$$C_{Fu} = \frac{1}{DT} \ln \left(\frac{Y_{fN}}{Y_{f0}} \right) + \frac{Y_{A0}}{Y_f} \frac{ke}{\rho_i} e^{-\frac{E}{RT_0}}$$

$$A_{Fu} = \frac{\frac{1}{DT} \ln \left(\frac{Y_{fN}}{Y_{f0}} \right) + \frac{E_2}{Y_f} (Y_{Fu(i+1)} + Y_{Fu(i-1)} - 2Y_{Fu(i)})}{C_{Fu}}$$

$$Y_{AN} = \text{STC} (\xi_N + Y_{FuN})$$

Note: When reaction rates are small, this reduces to the regular finite difference approximation. Also, when chemical reaction is slow but evaporation rate is high, the above equation yields correct solution.

For $\xi_N < 0$

Y_A limits reaction rate

Assume $Y_A = A_A + B_A e^{-C_A t}$

in
$$\frac{\partial Y_A}{\partial t} = \frac{E_2}{\bar{Y}_f} (\phi Y_{A_0}) - Y_A \left[\frac{STC \cdot Y_{Fu_0} k e^{-\frac{E}{RT_0}}}{\bar{Y}_f \rho_i} + \left(\frac{1}{\bar{Y}_f} \frac{\partial Y_f}{\partial t} \right) \right]$$

$$C_A = \frac{1}{DT} \ln \frac{Y_{fN}}{Y_{f_0}} + \frac{STC \cdot Y_{Fu_0} k e^{-\frac{E}{RT_0}}}{\bar{Y}_f - \rho_i}$$

$$A_A = \frac{\frac{E_2}{\bar{Y}_f} [Y_{A_0} (i+1) + Y_{A_0} (i-1) - 2Y_{A_0} (i)]}{C_A}$$

$$Y_{AN} = Y_{A_0} e^{-C_A DT} + A_A (1 - e^{-C_A DT})$$

$$Y_{FuN} = \frac{Y_{AN}}{STC} - \xi_N$$

Again, for small reaction rates and evaporation rate, the solution reduces to regular finite difference.

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